FTD-MT-24-89-70

FOREIGN TECHNOLOGY DIVISION

AIR FORCE SYSTEMS COMMAND

USE OF ADDITIVES FOR LUBRICATION OF INDUSTRIAL EQUIPMENT

by

A. M. Kuliyev, F. G. Suleymanova, and I. I. El'ovich

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EDITED MACHINE TRANSLATION

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English pages: 206

This document is a Mark II machine aided translation, post-edited for technical accuracy by: Charles T. Ostertag.

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UR/0000-69-000-000

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U. S. BOARD ON GEOGRAPHIC NAMES TRANSLITERATION SYSTEM

Block	Italic	Transliteration	Block	Italic	Transliteration
А а	*А а*	A, a	Р р	*Р р*	R, r
Б б	*Б б*	B, b	С с	*С с*	S, s
В в	*В в*	V, v	Т т	*Т т*	T, t
Г г	*Г г*	G, g	У у	*У у*	U, u
Д д	*Д д*	D, d	Ф ф	*Ф ф*	F, f
Е е	*Е е*	Ye, ye; E, e*	Х х	*Х х*	Kh, kh
Ж ж	*Ж ж*	Zh, zh	Ц ц	*Ц ц*	Ts, ts
З з	*З з*	Z, z	Ч ч	*Ч ч*	Ch, ch
И и	*И и*	I, i	Ш ш	*Ш ш*	Sh, sh
Й й	*Й й*	Y, y	Щ щ	*Щ щ*	Shch, shch
К к	*К к*	K, k	Ъ ъ	*Ъ ъ*	"
Л л	*Л л*	L, l	Ы ы	*Ы ы*	Y, y
М м	*М м*	M, m	Ь ь	*Ь ь*	'
Н н	*Н н*	N, n	Э э	*Э э*	E, e
О о	*О о*	O, o	Ю ю	*Ю ю*	Yu, yu
П п	*П п*	P, p	Я я	*Я я*	Ya, ya

* ye initially, after vowels, and after ъ, ь; e elsewhere. When written as ё in Russian, transliterate as yё or ё. The use of diacritical marks is preferred, but such marks may be omitted when expediency dictates.

FOLLOWING ARE THE CORRESPONDING RUSSIAN AND ENGLISH

DESIGNATIONS OF THE TRIGONOMETRIC FUNCTIONS

Russian	English
sin	sin
cos	cos
tg	tan
ctg	cot
sec	sec
cosec	csc
sh	sinh
ch	cosh
th	tanh
cth	coth
sch	sech
csch	csch
arc sin	$\sin^{-1}$
arc cos	$\cos^{-1}$
arc tg	$\tan^{-1}$
arc ctg	$\cot^{-1}$
arc sec	$\sec^{-1}$
arc cosec	$\csc^{-1}$
arc sh	$\sinh^{-1}$
arc ch	$\cosh^{-1}$
arc th	$\tanh^{-1}$
arc cth	$\coth^{-1}$
arc sch	sech^{-1}
arc csch	csch^{-1}
rot	curl
lg	log

INTRODUCTION

The development of research works in the field of synthesis and production of oils additives created conditions for the radical improving of quality of lubricating oils and made it possible to change these qualities in the required directions. The book covers the basic principles for determination of the necessary level of alloying, proceeding from an analysis of conditions of operation of the basic forms of industrial equipment and their requirements for the quality of additives for lubricating oils.

For a rational organization of investigations in the field of selection of additives for the basic types of industrial equipment the machinery and mechanisms are classified by structural, kinematic, and dynamic factors. An analysis of efficiency based on maximum states of kinematic pairs makes it possible to determine the basic directions and required level of alloying under specific conditions. With the help of the complex of preliminary methods of sampling and tests of additives on laboratory machines, instruments, and installations, including an appraisal of functional properties of additives to oils (antiwear, antipitting, antiburr properties, stability, corrosion, depolymerized stability, and others), and also a complex of methods of bench tests, the appraisal of the most important operational qualities of additives is ensured.

This experimental material characterizes the use of a complex of test methods during the development of requirements for the quality of additives. On the basis of the equipment and methods developed a wide assortment of additives is investigated which were synthesized during the last few years at the IKhP of the Azerbaydzhan SSR Academy of Sciences under the leadership of A. M. Kuliyev and which were utilized as effective additives for improving the operational qualities of oils for the lubrication of mechanisms of industrial equipment and transport. Investigations are made of the basic functional properties of IKhP additives and compositions based on them.

Results are cited from industrial and performance tests of additives and their compositions under full-scale conditions in a number of basic branches of industry (heavily loaded toothed and worm transmissions, used in reduction gears for industrial equipment, ship mechanisms, face packing of electric drills, gear boxes and main reduction gears for automotive transmissions, certain special types of equipment).

On the basis of analysis of the effectiveness of application of additives to oils a method has been developed for appraisal of economic factors of their application, and also paths are outlined for the standardization of nomenclature of industrial oils on a base of a complex of additives developed at the IKhP of the Azerbaydzhan SSR Academy of Sciences.

CHAPTER I

PRESENT STATE AND PROSPECTS OF DEVELOPMENT OF PRODUCTION OF LUBRICATING OILS FOR INDUSTRIAL EQUIPMENT

In solving the problem of development of public production of the country, Soviet industry during the period from 1959 through 1965 increased the volume of output by 1.84 times, and for the five-year period 1966-1970 in accordance with directives of the Twenty-Third Congress of the CPSU an increase in volume of industrial production by 1.5 times is projected. The five-year plan for the development of the national economy of the USSR anticipates an acceleration of growth of labor productivity in all branches of production, mastering the output of new improved and high-quality articles, and maximum intensification of all branches of production. Here the mission is "to accelerate scientific and technical progress on the basis of the wide development of scientific investigatings and the rapid use of their results in production ..."

The solution of this problem is ensured, in particular, by the development "... of investigations in the field of chemistry for the development of new economically profitable chemical processes and obtaining of effective substances and materials."

In recent years wide development has been achieved by research works in the field of synthesis, production, testing and application of new means for improvement of the qualities of lubricating oils with the help of special additives – allowing components for lubricants

which change their operational properties in the required directions. The tremendous influence which is exerted on the development of industrial production by the introduction in the national economy of highly effective lubricating materials, alloyed by special additives, makes the problem of development of scientifically proved methods of allowing of lubricating oils one of the basic directions of scientific and technical progress.

Scientific research works in the field of development of new samples of alloyed oils and introduction of the results in many branches of industry in the Azerbaydzhan SSR showed a significant increase of productivity and economy of machines, a considerable increase of their reliability and life, and an increase of engine life by 1.5-2 times.

The considerable economic effect from the application of lubricating oils with additives in the national economy conditioned the necessity, specified by the directives of the Twenty-Third Party Congress, "to use effective additives in the production of all diesel automotive oils." However, this group of lubricating oils makes up only a part of the overall volume of all forms of industrial oils produced in the USSR. The concept of the requirements of the national economy for lubricating oils in the immediate future is characterized by the following data (T, millions):

Group of oils	1970	1975	1980
Motor	4.8	5.7	6.5
Transmission	0.3	0.4	0.5
Industrial	3.3	4.0	4.7
All told	8.4	10.1	11.7

The considerable success, attained in the field of alloying of motor oils, made it possible to extend the investigations also in the direction of improvement of the operational characteristics of a large number of technical oils used in various areas of contemporary technology.

One of the most urgent problems of contemporary technology is the solution of the problem of increasing the reliability and longevity of machines. On a national scale a great deal of resources are expended for these goals. It is sufficient to say that only for repair more than 15 billion rubles are expended yearly, over 2 million persons work in repair services, and around 800 thousand metal-cutting machines are tied up.

However, in the field of development of theoretical and practical bases for the alloying of industrial oils very little has been done. In spite of the significant expansion of the assortment of additives of different assignment, in the designing and assignment of technical conditions for the exploitation of contemporary machines and mechanisms most frequently obsolete principles of selection of lubricating materials are followed – from the period when lubricating oils were used without additives. This promoted the circumstance that selection of oils for a specific machine was determined not by objective conditions of its exploitation, but by the existing classification of oils based on their areas of application.

For example, a classification of oils exists which is based on the primary areas of their application, according to which all mineral lubricating oils are split into the following basic groups:

1) industrial oils, utilized for lubrication of mechanisms of equipment of different branches of industry and basically characterized by a universality of assignment;

2) oils for internal-combustion engines;

3) transmission oils, used mainly for lubrication of power units of transmissions (chiefly transport machinery);

4) cylinder oils, used for lubrication of piston steam engines;

5) oils used for lubrication of water and steam turbines, generators of electrical current and control systems (turbine oils);

6) oils for the lubrication of air compressors, blowers, and compressors of refrigerating machinery (compressor oils and oils for refrigerating machinery);

7) instrument oils for lubrication of different mechanisms, instruments, and apparatuses;

8) oils for lubrication of special mechanisms – ship, axial, and others.

The assortment of mineral oils also includes insulating, hydraulic, shock absorbing and other sorts.

For all these groups, in accordance with effective recommendations on the selection of lubricating oil for machines and mechanisms calculation of the viscosity of oil, ensuring liquid friction is made mainly stemming from the hydrodynamic theory of lubrication. Thus, for journal bearings the necessary lubricating material is selected, for example, by the Fal'ts formula:

$$\eta = \frac{P}{3380000\, d^{3.5} \frac{l}{d}\, n},$$

where η – coefficient of dynamic viscosity, kg·s/m^2; P – overall load on bearing, kg; d – diameter of bearing journal, m; l – length of bearing, m; n – number of turns of shaft per minute.

For the selected lubricating oil a verifying calculation of the temperature of the lubricating layer by the formula:

$$T = t + \sqrt[2.6]{\frac{P\, n^3 \eta}{9600\, d^2 \frac{l}{d}}},$$

where t – temperature of surrounding air, °C, a – radiation factor.

For lubricating oils, used in anti-friction bearings, the basic index of quality is also the viscosity of temperature of dropping (for grease lubricants). Selection of viscosity of lubricating oil is made depending on the dimensions of the bearing, number of turns, and operating temperature. The sorts of mineral oils recommended for this purpose are: oil for high-speed mechanisms, separator, industrial-12, industrial-20, industrial-30, industrial-45, cylinder-11, AK-10, AK-15, P-28.

For flat sliding surfaces (guides for reciprocating shifting of components and subassemblies) the oils are selected depending on loads and rates of slip. The determining parameter here is also the viscosity of the oil.

The following sorts of oils are recommended for lubrication of guides [1]: industrial-20, industrial-30, industrial-45, AK-10, AK-15 and cylinder-11; for lubrication of toothed and work transmissions in reduction gears – industrial-30, industrial-45, AK-10, cylinder-11; AK-15, nigrol cylinder oil GOST [All Union Government Standard] 542-50, P-28 (bright stock), cylinder-38 (cylinder-6). Selection is made also based on viscosity, proceeding from the value of specific loads, rate of slip, and average temperature of surrounding air.

Thus operational recommendations practically do not consider the possibility of radical improvement of the quality of oils with the help of alloying – i.e., introduction into their composition of additives which change the properties of oils in assigned directions.

Up to now the basic trend of scientific-research and experimental works in the area of selection of additives for industrial oils. But together with the continuous expansion of the area of application of additives the assortment of them is increasing. It turns out (and this is confirmed by operational experience with industrial oils in the most diverse fields of their application) that the same additive can

satisfy conditions of operation of machines, mechanisms, subassemblies, and units, where based on conventional classification of industrial oils it is anticipated to use their own oils, specially selected for the particular class. Such additives are, for example, IZ-23k and DF-11, which find application both in motor and also in machine oils, and the additives ionol, used in transmission, power turbine, and hydraulic oils, and many others.

The practice of application of oils with additives also shows that as oils of different classes the same commercial products can be used. In particular, light industrial oils, related by conventional classification to machine oils, find application as motor, machine, hydraulic, etc.; diesel oils (class of motor oils) are used as machine oils.

Thus it becomes evident that development and creation of an assortment of industrial oils and to a still greater degree the selection of additives for industrial oils of different assignment should be more rationally based on definite objective factors, considering the conditions of operation of a specific mechanism of machine, requirements of operational reliability, and longevity. The intensity of influence on these factors, which is determined by conditions of production and exploitation of the machine and is realized by the introduction of additives into the composition of the lubricant, should characterize the necessary level of alloying for the given conditions.

CHAPTER II

BASIC TRENDS IN ALLOYING OF INDUSTRIAL OILS

Alloying of lubricants should be considered one of the basic factors influencing the longevity and reliability of machines and mechanisms. Here it is possible to assume that a lubricant should be considered a structural material.

Proceeding from this it is necessary to establish basic indices, which can characterize the reliable and enduring operation of machines and mechanisms with the use of alloyed oils and basic trends and means of derivation of physicochemical, mechanical-chemical, and physico-mechanical properties of alloying components in accordance with the requirements determined by conditions of operation of the lubricant under assigned conditions.

In spite of the evident urgency and importance of the problem of longevity and reliability of machines on the whole, its basic positions were determined relatively recently (1934) [2]. Subsequently it has been developed by the works of Soviet scientists in the direction of practical use in the basic branches of machine building. The fundamental works of V. N. Kuznetsov, Yu. A, Ishlinskiy, B. V. Deryagin, P. A. Rebinder, M. M. Khrushchov, B. V. Kostetskiy, I. V. Kragel'skiy, A. S. Akhmatov, and others created the theoretical base for connecting the basic questions of the problem of longevity and reliability with conditions of operation of elements of mechanisms and machines.

Since 1939 four All-Union Conferences on friction and wear in machines [3-7], and also a number of conferences on theory of friction and wear, and special conferences on longevity and reliability summarized the achievements of various Soviet schools in this area of science. Starting with the Second All-Union Conference the new scientific trend finds reflection in the works of these conferences. Its basic content is the development of bases for changing the functional properties of lubricating oils with the help of additives (alloying of oils).

Let us establish the basic ideas of longevity and reliability of machines in reference to conditions, when as the main factor affecting these indices of machines we will take these or other influences of alloyed lubricants, conditionally examined as structural materials.

Let us assume that fulfillment of working functions of a machine or mechanism is determined beforehand by assigned requirements of accuracy of interaction or is limited by indices of operational effectiveness and economic expediency of use for a definite period - the period of unfailing operation of the machine. Then the reliability of a given mechanism or machine will be the property, evaluated by the probability of fulfillment of its intended working function during a definite period of service (with assigned properties of lubricant used). By longevity we understand the property of a machine to fulfill its working function for a specific period of service with a previously specified probability of premature breakdown of one or several components (for reasons connected with the different effectiveness of the lubricant), making up the kinematic chain of basic or auxiliary mechanisms of the particular machine.

For every machine, based on average statistical data, it is possible to establish a certain medium interval between alternate failures in operation from damages or breakdown of conjugate machine parts due to insufficient effectiveness of the lubricant material used. On the whole one or another lubricant under assigned conditions of

exploitation ensures a certain efficiency of the machine, i.e., its ability to fulfill a working function with previously assigned technical and economic conditions during a definite period of service.

In general when considering the influence of lubrication on the limits of efficiency of machines, one may assume that the most general and important factors are temperature phenomena and the combined various forms of influences, and also changes in structures and voltage in the surface layers of elements of kinematic pairs. These general criteria are manifested especially brightly in that influence which alloying of a lubricant exerts on such causes of loss of efficiency as fatigue breakdown, wear, disturbance of conditions contact of components due to thermal instability of lubrication, etc.

Thus exhausting of efficiency of machine can be examined only as a result of the interaction of elements of kinematic pairs under assigned working conditions with specific properties of lubricating materials. When considering the influence of lubricating material it is important to determine correctly the maximum conditions, i.e., after achievement of which efficiency will be recognized as exhaused. It is naturally that based on their physical, physicochemical, and physicomechanical essence these maximum states can be essentially different and each of them can have a corresponding one or several specific functional properties of lubricating material.

As will be clear subsequently, in reality these bonds are extraordinarily complex, sometimes even simply contradictory. Nevertheless it is namely they which make up the basis of influence on longevity and reliability of machines by this or that change in the functional properties of a lubricant, mainly by its alloying.

The most important forms of maximum states of machines, on which the quality of lubricating material and the functional properties of alloying components exert a significant influence, are: the appearance of maximum clearances; appearance of criteria of maximum damage due to contact fatigue; corrosional damage; increase of operating temperature,

leading to abnormal interaction or being the result of incorrect conditions of coupling of parts; loss of vacuum seal; maximum dynamic loads due to vibrations, unbalance, etc.

A more detailed description of characteristic maximum conditions can be gained in an analysis of conditions of operation of elements of kinematic pairs of the particular machine under concrete conditions of work.

1. Influence of Additives on Change of Maximum Conditions of Mechanisms of Machinery

A number of basic forms of influences exist which cause a gradual loss in the efficiency of machine parts. The intensity and direction of development of the majority of these can be influenced by a change in the operational properties of the lubricating composition by the addition of additives. Certain trends of this influence are examined below.

With Variable Stresses

The modern theory of strength of metals has been developed fruitfully on the basis of basic positions of the theory of dislocations [8]. In particular, the theory of fatigue breakdown of metals developed by N. S. Akulov and V. A. Franyuk [9] defines the basic stages of fatigue breakdown. Ordinary dislocation lines (up to 10,000 lattice spacings[1]), leading to formation of groups of dislocations, promote an increase in state of stress. Following this a breakdown of dislocations leads to the formation of microscopic cracks, which in turn lead to the formation of fatigue cracks.

The properties of quasi-rigid boundary lubricating layers which are alloyed by additives, as we will see from the subsequent analysis,

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[1]Lattice spacing — 3-6 Å.

essentially influence the values of normal and tangential stresses in the surface layer. On the one hand this exerts a direct influence on tensile strength of the material of the surface layer, and on the other – will indirectly affect its structural, physicomechanical, chemical-mechanical, and physicochemical modification.

As it is known, stress cycles in the case of variable influence are characterized by:

amplitude:

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2};$$

coefficient of asymmetry of cycle:

$$r = \frac{\sigma_{min}}{\sigma_{max}};$$

average stress of cycle:

$$\sigma_{cp} = \frac{\sigma_{max} + \sigma_{min}}{2}.$$

It is simple to see that

$$\sigma_a \cdot \sigma_{cp} = \frac{\sigma^2_{max}}{4}(1 - r^2).$$

It is known that the value of variable stresses and number of stress cycles up to the appearance of fatigue are connected by a dependence, presented in a general form by a hyperbola of a higher order.

The "canonical" law of fatigue is expressed by the following equation [10]:

$$\sigma_{-1N} = \frac{A_0}{N_u^{1:m}} = \sigma_{-1}\left(\frac{N_Б}{N_u}\right)^{1:m},$$

where σ_{-1N} – particular fatigue limit, i.e., that stress which the material can sustain up to a number of cycles $N_{\text{Ч}}$; σ_{-1} – fatigue limit during a symmetric variable cycle with a base number of cycles $N_{\text{Б}}$; $N_{\text{Ч}}$ – median number of cycles of a particular fatigue limit; $N_{\text{Б}}$ – base median number of cycles of the basic fatigue limit (usually for steel $N_{\text{Б}} = 10^7$, for nonferrous metals $N_{\text{Б}} = 10^8$); A_0, m – constants, depending on properties of the materials.

As experimental data show, leaving a constant value σ_{-1N} it turns out to be possible, depending on the chemical structure and concentration of alloying agents introduced in the lubricating lay, which is found under conditions of surface contact, to increase $N_{\text{Ч}}$.

We will examine the basic directions, by which it is possible to influence a change of fatigue longevity by a change in the physicochemical properties of the lubricating layers.

Investigations [11-14] makes it possible to establish that the properties of the boundary lubricating layers, depending on their physicochemical characteristics, can exert an influence on the following factors, connected directly with fatigue longevity of the material of surface layers: temperature in zone of contact, geometry of zone of contact, thickness of lubricating layer, and extent of internal friction.

By changing the physicochemical properties of boundary lubricating layers due to adding alloying components to them it is possible to intentionally change these properties for the purpose of creating conditions which are favorable for increasing contact fatigue longevity.

Let us consider the influence of physicochemical properties of lubricants on changing factors which have an effect on longevity during contact fatigue.

Change of microgeometry of contact. Deformations and changes of dimensions of contacting bodies due to susceptibility to wear, occurring in connection with specific properties of the lubricating layer, lead

to a change in the values of the given radii of curvature and, consequently, in the values of contact stresses. It is possible also to point out changes in the mechanism of free play.

As is known, the dependences established by Hertz are correct for static conditions. Real conditions of dynamic contact are different in many respects from the conditions examined by classical theory of Hertz. The properties of a lubricating layer, determining its behavior under conditions of dynamic contact under the assumption that the Newtonian nature of the liquid is preserved, are established readily from known relationships. On the other hand, analyzing the distribution of stresses on elastic contact under the assumption that the boundary lubricating layer possesses the properties of a "quasi-rigid" body, it is possible to establish a direct connection between elastic constants of the lubricating layer and maximum carrying capacity of the contacting surfaces.

The effect of a change of lubricating capacity of an oil film under conditions of elastic contact was investigated by Khasimoto [15]. Usually during an analysis of conditions of work of a lubricating layer in the clearances between surfaces which are rolled over each other with finite values of curvature (model of contact of working profiles of teeth of a gear drive and rolling bodies of antifriction bearings) they originate from theory of elastic deformation of contacting surfaces (theory of Hertz). The classic theory of Hertz determines only the value of stresses on static contact under the impact of a normal load The theory of Hertz, however, absolutely does not consider the presence on contact of an oil film possessing specific lubricating properties. This film, which is found in the clearance between contacting surfaces, prevents the direct contact of metallic surfaces.

Contemporary theoretical concepts concerning the nature of distribution and absolute values of tangential stresses in points which are located under the contact surface [16-18] give certain possibilities for the calculation of the influence of changes of conditions of contact on the values of main stresses, which determine the maximum states of contacting surfaces.

Let us give certain data [19], characterizing the contact picture for a case of rolling of toroid rollers on cylindrical (Table 1).

Table 1. Stresses and deformations during contact.

Radius of roller, mm	Maximum stress of shift, S_s kg/mm²	Semiaxis of ellipse of contact, mm		$\frac{S_0}{S_s}$ [18]	$\frac{b_0}{a}$ [18]
		$a \times 10^{-3}$	$b \times 10^{-3}$		
6,35	$23{,}5\sqrt[3]{P_0}$	$94\sqrt[3]{P_0}$	$70\sqrt[3]{P_0}$	1,20	0,31
9,75	$19\sqrt[3]{P_0}$	$87\sqrt[3]{P_0}$	$87\sqrt[3]{P_0}$	1,35	0,35
12,7	$18{,}2\sqrt[3]{P_0}$	$84\sqrt[3]{P_0}$	$100\sqrt[3]{P_0}$	1,39	0,37

On the other hand, theories exist (Martin and others) in which only the effect of the presence of an oil film with specific lubricating properties between contacting surfaces is examined. The deficiency of these theories lies in the fact that they do not consider elastic deformation of contacting surfaces.

Thus the solution of the problem in first approximation (disregarding various side phenomena, conditioned by complex dependences of physicochemical properties of the lubricating layer on conditions of contact) consists of an analysis of the behavior of an oil film with specific lubricating properties between surfaces which are in contact and possess finite values of elastic properties.

Khasimoto's solution consists of a unification of integral equations of elastic deformation, obtained according to the theory of Hertz, with differential equations of the oil film, obtained proceeding from the theory of Martin.

n.ine

Using for the analysis of the oil layer the Navier-Stokes equation for a constant two-dimensional flow of incompressible liquid, in which we disregard members considering inertia and take dimensions of oil layer small as compared to the radii of curvature, we obtain:

$$\frac{1}{12\mu}\frac{dp}{dx}=\frac{Q}{h^3}-\frac{U}{h^2}, \tag{1}$$

where μ — coefficient of dynamic viscosity, p — pressure, x — current coordinate, Q -- expenditure of liquid, h — distance between rolled surfaces with an assigned value of current coordinate x;

$$U=\frac{U_1+U_2}{2},$$

where U_1 and U_2 -- peripheral velocities of rolled cylindrical surfaces.

For thickness h of the oil film between two rolled cylindrical surfaces with radii of curvature R_1 and R_2 the following analytic dependence is correct

$$h=\frac{x^2}{2R}-\frac{4}{\pi E}\int_{-l_1}^{l_2}p(r)\log|r-x|dr+H, \tag{2}$$

where R — given radius of curvature, E — elastic modulus, H — constant.

Thus during resolution by analysis of the problem of the state of an oil film taking into account elastic deformation of rolled surfaces it is sufficient to unite and jointly solve equations (1) and (2).

Considering that $p=\frac{dr}{dx}=0$ with $x=-l_1$ and $p=0$ with $x=l_2$ it is obvious that the effective limits of existence of the carrying lubricating layer is region, limited by

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$$-l_1 \leqslant x < l_2.$$

When solving the problem, accepting

$$h = \frac{x^2}{2R} + h_{min}$$

and not considering elastic deformation of rolled surfaces, it is possible to use the method of Martin. Formula (1) is written in the form

$$\frac{1}{12\mu}\frac{dp}{dx} = \frac{Q}{\left(\frac{x^2}{2R} + h_{min}\right)^3} - \frac{U}{\left(\frac{x^2}{2R} + h_{min}\right)^2}$$

Integrating for x and taking $l_2 = \infty$, we obtain

$$p\frac{h_{min}^2}{12\mu} = U\sqrt{2Rh_{min}}C.$$

The numerical solutions obtained by Khasimoto with 2 < U < 600 cm/s and pressures up to 10,000 kg/cm^2 are characterized graphically by the curves in Fig. 1.

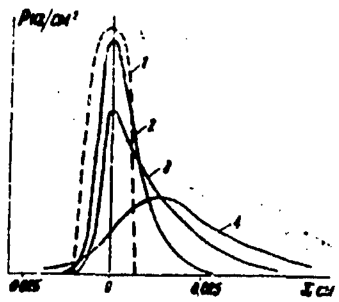


Fig. 1. Relationship between dimensions of contact by pressure and speed. 1 - according to the theory of Hertz; at the speed of rolling: 2 - 2 cm/s; 3 - 30 cm/s; 4 - 600 cm/s.

With large values of U the influence of elastic deformation is insignificant and the results approximate those obtained according to the theory of Martin. Maximum pressure has the greatest value at U = o.

The value of maximum pressure, calculated according to Hertz and not depending on speed, characterizes the upper boundary of the region in which p are disposed depending on speed.

From equation (2) it follows that other things being equal with an increase of constant of elasticity E there is an increase in the thickness of the oil film dividing contacting surfaces. Together with an increase of the latter maximum contact pressures will decrease and, consequently, contact-fatigue longevity is increased.

A considerable influence is exerted on the longevity of contacting surfaces by the rheological properties of the lubricant which are regulated in wide limits by additives.

Thickness of the lubricating layer. Investigation of the lubricating film in the spaces between contacting surfaces if the object of numerous investigations [20-26]. Lane and Hughes [20] carried out experiments with the help of direct measurements of values of resistance on contact with a current of 0.5 A. Crook [21] determined the thickness of a film with the help of measurement of the amount of oil passing through the zone of contact. The value of thickness of the lubricating layer was of an order of 2 μm. Lewicki [22], using the capacity method and Brix [23], using the method of voltage drop, estimate the values of thickness of a lubricating film of an order of tens of μm. Thus, according to Brix, for thrust and radial bearings the thickness of the lubricating film varies within limits from 2.5 to 25 μm.

Twias [27] found that under conditions of pure sliding the thickness of the lubricating film turns out to be considerably less than during rolling (it changed from 0.075 to 0.25 μm).

Thus hydrodynamic effects, ensuring an increase in the thickness of the lubricating layer, are strengthened during transition from slipping to rolling.

Zuidema [25] points out that conditions of hydrodynamic lubrication, promoting the formation of a stable carrying layer, appear with clearances greater than 0.6 μm.

In experiments on disks [28] with a pressure of around 30 kg/cm the thickness of the lubricating film comprised around 3 μm.

Kagan, Bogdanov, and Yantovskiy [29] in the case of loading a thrust bearing obtained a thickness of lubricating layer of around 150 μm.

Cameron and others [30-32] investigated the thickness of lubricating film during the operation of steel and cast-iron cogwheels and found it equal to approximately one μm, and the minimum value pertains to sections of working profile which are found on the dividing circumference. In experiments, conducted with a constant load of 36 kg/cm and using SAE-30 oil a lowering of the thickness of the film was observed with an increase of temperature and corresponding lowering of viscosity of the oil. On the section of profile which corresponds to the location of the dividing circumference the thickness of the lubricating film turned out to be equal to 2 μm, and at the roots - 500 Å.

Thus under conditions of pure rolling the thickness of the lubricating film is higher than with a combination of rolling with slipping. Furthermore, the thickness of film, sufficient for formation of durable polymolecular layers with the properties of a "quasi-rigid" body, promotes an effective influence on the nature of distribution and absolute values of dynamic loads in the contact zone. Direct measurements, the results of which are given in [31], show that the thickness of the lubricating film does not exceed 10 μm.

In appraising the values of thickness of lubricating film with values up to 10 μm, we arrive at the conclusion that these values are commensurate with the unevenness of the surface and dimensions of foreign particles in the lubricant. Consequently the area of preparation

of conclusions concerning the nature of influence of thickness of lubricating layer, even one possessing definite "elastic" properties, is somewhat limited due to the influence of outside factors — the presence of third bodies in the spaces between contacting surfaces.

Internal and external friction. Speranskiy [5] in his theory of the interconnection of internal friction and thermogeneration during external friction shows that the temperature of the surface layers of rubbing parts increases with a larger value of damping decrement of oscillations (measure of internal friction).

However, the interconnection of phenomena of thermogeneration under different conditions of internal and external friction is influenced significantly by surface phenomena connected with the presence of adsorption interaction of the lubricant with surfaces of rubbing machine parts.

The chemical bond of the polar groups of additives, which were introduced in the boundary lubricating layer, with atoms of metal strengthens the adsorption interaction and changes depending on the chemical nature of the additives used.

It is known [33, 34] that strength of the bond of the boundary lubricating layer with the surface of the metal can be commensurate with the strength of the bond in solids. Under specific critical conditions, however, the presence of a surface active lubricant on friction surfaces, as this is shown by P. A. Rebinder and N. N. Petrova [35], causes surface dispersion of the metal. For example, the absence of pitting when using oils without additives testifies to the significant value of the dispersive effect of surface lubricating layers which are activated by specific hydrocarbons. Furthermore, it has been established [36] that thermal stability and antioxidant stability of a lubricating material has a significant effect on the reliability of work of antifriction bearings based on the criterion of maximum contact fatigue longevity. Consequently, by influencing the thermal stability and oxidizability of a lubricant, we also

indirectly influence the longevity of work of contacting surfaces at the case of their fatigue breakdowns due to the action of variable stresses.

Certain authors cite findings about the negative influence of chemically active additives, in particular those intended for application under severe conditions of operation (EP), on the maximum efficiency of contacting surfaces based on criterion of chipping. It is possible, for example, to cite the data of Scott [37], which are given in Table 2. As one can see, the most active additives in the region of superhigh pressures (chloroparaffin and dibutylphosphite), when taken in considerable concentrations (up to 10%), sharply lower the time up to the onset of fatigue breakdowns.

Table 2. Influence of additives on pitting of balls (load on upper ball 600 kg, n = 1500 r/min).

Sample	Time to breakdown, min
Oil without additive	41
Dibutylphosphite 1%	42
10%	8
Tricresylphosphates 1%	40
10%	43
Elementary sulfur, 0.4%	58
The same + lead naphthenate, 10%	50
Cnloroparaffin 1%	35
6%	14
10%	6
Graphite 1%	109
MoS_2 1%	79

However one should consider the difference of conditions of laboratory tests, in which these data were obtained, from some of the operating factors which are characteristic for actual conditions of exploitation. It is known, for example, that oils containing EP-additives effectively improve the conditions of work-in of surfaces. Consequently it is necessary to expect best conditions of distribution

of load between contacting surfaces in the case when additives of such a kind are used.

Redistribution of load on contact undoubtedly will have an effect on the phenomenon of chipping. Certain investigations [37] have established that the break-in of cogwheels on oil with chemically active additives changes ultimate load corresponding to the onset of fatigue breakdowns.

It has been shown [38] that the best polishing of surfaces promotes an increase of resistance to chipping.

Ye. Zaretskiy and his collaborators [36] also cite data about the influence of application of additives to lubricating oils on value of limit of contact fatigue longevity. In Table 3 characteristics are given for lubricating compositions which were used during tests on contact fatigue longevity, and in Table 4 – the results of these tests.

As can be seen from these data, the greatest lowering of contact fatigue longevity was observed when using a lubricating composition on a base of synthetic oils with inhibitors of oxidation.

Attention is merited, however, by the fact that oils are compared which are based on level of initial viscosity. This undoubtedly introduces significant corrections during the transfer of results obtained into actual conditions of influence of additives on fatigue longevity during elastic contact. Works are known in which a direct bond is established between viscosity of the lubricant and chipping. Thus Newman [39] showed that the viscosity of a lubricant has a considerable influence on chipping. At low viscosity the lubricant promoted a more intensive chipping that at high.

Experiments conducted on steel balls [40, 41] showed that the longevity of the balls is connected with the viscosity of the lubricating oil to the degree 0.2. For example, with a viscosity of

5.1 cSt at 100°F (37, 38°C) the average longevity comprised $4.0 \cdot 10^6$ cycles; under the same conditions when oil with a viscosity of 119.1 cSt was used the longevity was increased up to $9 \cdot 10^6$ cycles. If, however, one considers that with an increase of values of torque and number of turns an intensive increase of wear is observed [42], then during the appraisal of influence of lubricant on maximum state, determined by wear or fatigue breakdowns, one should consider the interaction of these forms of surface breakdowns and the influence of the resulting effect on the final condition of the surface.

Table 3. Condensed characteristics of tested samples of oils.

Lubricating material			Content of additives		Neutralization number before tests	Viscosity (cSt) at		
Kind	Make	Composition	Type	Concentration, %		37,8°	98,9°	148,9°
Esters	NA-XL-3	Ester of octyl alcohol, adipic acid and polyethylene glycol	Inhibitor of oxidation	0,5	0,18	35,30	7 14	3,40
	NA-XL-8	Di-2-ethylene xylol ester of sebacic acid	Inhibitor of oxidation Antiburr component Antifoam additive	0,5 5,0 0,0005	0,25	13,29	3,51	1.86
Mineral oil	NA-XL-4	Paraffin 63% Naphthenic 32% Aromatic 5%	Antiburr component	2,0	0,09	126,60	11,72	4,3
	NA-XL-7	Paraffin 66% Naphthenic 33% Aromatic 1%	Without additives	0	0,1	238,70	20,68	7,00

Thus, if one were to originate from above described effects, it is possible to expect that with using alloying agents in a lubricating composition which is in the boundary layers it is possible to increase the value σ_{-1}, which is equivalent to the effect of improvement of "structural" properties of the lubricating layer.

Table 4. Contact fatigue longevity of samples depending on quality of lubricant.

Tested lubricating material	Test on 5-ball machine at 150°		Test of bearing 7258 at 150°	
	Carrying capacity at $N = 1 \cdot 10^6$, kg	Relative carrying capacity	Carrying capacity at $N = 1 \cdot 10^6$, kg	Relative carrying capacity
NA-XL-3	370	0,68	2160	0,69
NA-XL-8	382	0,70	2340	0,75
NA-XL-4	540	0,98	2840	0,91
NA-XL-7	550	1 00	3100	1,00

With a Change of Maximum Clearances Due to Wear

The interaction of two contacting elements of kinematic pairs at definite speeds and loads is combined with phenomena of internal and external friction.

Processes, flowing both in the volume and also in the boundary lubricating layers, are found in direct dependence on the properties of the lubricating material.

Discrete contact of microroughnesses, leading to elastic and plastic deformations, and also to breakdowns of material and the accompanying development and break of intermolecular bonds, in the end causes wear of contacting surfaces. Phenomena of internal friction in the lubricating and surface layers of contacting materials can be regulated quantitatively and qualitatively by the introduction by alloying additives in the lubricating layer.

Irreversible losses of energy during friction accumulate mainly from dissipation of heat, hidden energy of plastic deformation, and losses by the change of surface energy.

The basic directions of action of alloying components for lubricants during friction and wear consist of the following.

During contact of two solids due to waviness and roughness of surface the actual area of contact is in constant dependence on external conditions and the mechanical properties of the surfaces. Elastic, and then also plastic deformation of microroughnesses in the process of friction can change the initial picture. Control of this process has important practical value in the selection of the optimum alloy for the lubricant. Discrete contact during slipping is accompanied by an increase of temperatures on spots of contact, which conditions the appearance of a gradient of mechanical properties.

Due to this change in process of friction of two bodies, a frictional bond appears at the points of contact which can be considered conditionally as a third body with modified mechanical and chemical properties. Under these conditions a decisive role is played by the action of additives which are introduced in the boundary lubricating layers.

As is known [43], friction has a dual molecular-mechanical nature. It is conditioned, first, by the surmounting of the adhesion bond between surfaces, most frequently covered by boundary films, and, secondly, by volume deformation. Thus, proceeding from the described mechanism of external friction, the greatest value from the point of view of possible influence of properties of surface layers, formed by alloying elements of the lubricant, is acquired by factors which influence their adhesional interaction and deformation.

Adhesion interaction. Adhesion is caused by forces of cohesion which operate between molecules and atoms and which decrease sharply with distance [44, 45]. The formation of a boundary film therefore hinders the development of adhesion directly between the rubbing surfaces. But these films themselves enter into an adhesion interaction, as a result of which conditions can take place in which the films break down.

In the case of destruction of the surface layers, when compression stress exceeds the strength of the surface layer, contact grasping of materials of the surface can take place [46].

According to the views of B. V. Deryagin and his school [47, 48], during contact a contact potential difference appears and the adjacent contacting surfaces can be identified with capacitor, the places of which is a double electrical layer.

Chemical activity of the lubricating layer in combination with the directed influence on the adhesion interaction of directly contacting surfaces and surface layers which are forming can therefore be important factors, on the change of which the process of friction will depend.

<u>Deformation of surface layers</u> with various chemical-mechanical changes, which can occur during the action of additives, will render an influence on the redistribution of regions with various micro-hardness. This influence naturally can be adjusted in the desired direction with the help of previously assigned properties of additives. The chemical activity of additives found in the zone of deformation of surface layers during friction can also be manifested in the following way. With an increase of temperatures on contact either a softening of surface layers can take place (if the temperature rate is stable) or a hardening of them (in the case of thermal flashes and rapid coolings). In both cases the influence of chemical properties of the layer is doubtless. They will appear also in the case of phenomena of metal wear due to various temperatures of softening, changing the intensity of the process and its direction.

Diffusion processes taking place in the surface layers also depend on the chemcial composition of the layer. According to Oding [49], the diffusion rate

$$V_D = -D_c \frac{\partial c}{\partial x} + D_s \frac{\partial s}{\partial x} + D_t \frac{\partial t}{\partial x},$$

where c – concentration, ε – deformation, t – temperature D_c – coefficient of diffusion, D_ε and D_t – proportionality factors.

The direction and rate of diffusion processes lead both to structural changes and to redistribution of active elements of the material and the surface layer, particularly in the presence of alloying additives.

The complex bond between chemical and thermomechanical phenomena in surface layers is supplemented by the effect, established by a number of authors, of an increase the capacity for oxidation of the surface layer which is subjected to plastic deformation [50, 51]. In the end the strength of the surface layers can, as was already indicated, be increased or decreased and naturally it is possible to control this change by artificially creating conditions for the desired direction of the process.

From the works of P. A. Rebinder [52] it is known that plastic deformation is facilitated by a lowering of free molecular forces appearing on surfaces. This lowering can take place if the surface is covered by adsorptive layers of surface-active substances. The activity of the lubricating layer here is obviously the main active principle.

Processes of generation and dissipation of heat. Above it was pointed out that many processes taking place in boundary layers are connected with the generation of heat in the zone of contact. From the usual presentations about distribution of generated heat between two bodies we have

$$Q_1 = \alpha Q,$$
$$Q_2 (1 - \alpha) Q,$$

where Q_1, Q_2 – heat, directed in bodies 1 and 2, α – coefficient of distribution of heat flows.

It is known [53] that

$$\alpha = \frac{\lambda_1}{\lambda_2}\sqrt{\frac{a_2}{a_1}},$$

where λ_1 and λ_2 – thermal conductivities, a_1 and a_2 – temperature diffusivities.

Furthermore a dependence is known where α is also connected with the rate of slip [54]

$$1-\alpha = \frac{\sqrt{\pi\sigma'}}{\sqrt{\pi\sigma'}+\sqrt{\rho c v}},$$

where σ' – coeffici nt of heat transfer, ρ – density, c – specific heat, v – rate of slip.

The actual state of the contacting surface at the time of application of load has a decisive influence on thermal conductivity of contact or on the value reverse to it – thermal resistance. The latter decreases with an increase of pressure and decrease of roughness (increase of actual area of contact).

With an increase of pressure the elastic deformation of micro-roughnesses causes an increase in the area of actual contact, which leads to an increase of its conductivity.

Investigations conducted by V. S. Shedrov [55] give an idea about the nature of change of the coefficient of distribution of heat flows on contact. It was determined that thermal resistance of contact, and also heat generation in it are influenced by boundary films [55]

$$\frac{\alpha}{1-\alpha} = \frac{q_1}{q_2}\cdot\frac{\left(\frac{h}{2\lambda_1}+\frac{l}{\lambda_2}\right)}{\frac{h}{2\lambda'}+\frac{l}{\lambda_1}},$$

here h – overall thickness of touching boundary films, λ' – coefficient of thermal conductivity of boundary lubricating layer, λ_1, λ_2 – coefficients of thermal conductivity of rubbing materials.

Thus the distribution of heat flows between friction bodies and the lubricating layer depends on the coefficients of thermal conductivity of materials of the surfaces, mutual covering, and expenditure of lubricant.

An analysis of the distribution of heat flows between friction surfaces, the film of the boundary lubricating layer, and the environment turns out to be very important for consideration of the many directions of the lubricating action of additives to oils.

Stability of boundary layers. Electron diffraction and diffraction of X-rays [56, 57, 58] show that regardless of whether the boundary layer is supported on the surface of the metal by chemical or physical forces or a combination of them a considerable orientation of polar molecules in the boundary layer takes place. Inasmuch as the influence of polar components on the coefficient of friction pertains to the class of surface phenomena, insignificant concentrations of these components are sufficient.

The first investigations in the field of additives to mineral oils for increasing their oiliness [59, 60, 61] and the first patent for the application of such additives (olenic acid) [62] opened an extraordinarily effective path for the use of surface-active substances in this direction.

Investigations [24, 63-65] of temperatures of phase transitions during the lubrication of steel with solutions of polar compounds, constituting hydrocarbons with long chains, showed that physically adsorbed layers cause a lowering of the coefficient of friction, and the effectiveness of condensed molecular layers in the section of their capacity for adsorption on metallic surfaces increases in the following sequence [66-69]: esters – alcohols – carboxylic acids – primary amines.

This sequence can be explained by the fact that energy of adsorption U_0 is directly proportional to the square of dipole moment m and inversely proportional to the cube of distance r from the electrical center of gravity of the molecule up to the adsorted surface

$$U_0 = \frac{km^2}{r^3}.$$

It follows from this that for every compound there is a specific temperature of disorientation or desorption of molecular layers. Intervals of temperatures comprise: for fatty alcohols 38-99°, for fatty acids 38-130°, for amines 99-150°.

Temperature regimen of the surface affects the disorientation of molecular chains and leads to the decomposition of the film of lubricant or promotes the formation (with the presence in the composition of the lubricant of chemically active components) of eutectic films, improving the conditions of friction.

According to Bowden and Tabor [70, 71] the temperature of disorientation coincides with the melting point of soap, formed as a result of the chemical reaction of a fatty acid with a metallic surface. As can be seen from the data in Table 5, temperatures within limits of 100-140° are the most characteristic for disorientation of a lubricant.

To ensure the protective role of the lubricant at higher temperatures chemically active additives are used which are effective up to 200-250°.

It is shown in works [72-76] that physical adsorption predominates over chemical adsorption; here the adsorption capacity of acids is higher than that of alcohols and esters, and the adsorbed film is monomolecular.

Table 5. Melting point of soaps.

Metals	Fatty acids				
	Lauric (C_{12}), m.p. 4[illegible]	Myristic (C_{14}), m.p. 53.[illegible]°	Palmitic (C_{16}), m.p. 6[illegible].[illegible]°	Stearic (C_{18}), m.p. [illegible].2°	Octanoic (C_{22}) m.p. 9[illegible].4°
Platinum	35	—	55	60	[illegible]
Silver	—	80	—	45	—
Copper	70	—	—	60	90
Steel	105	—	—	125	140
Zinc	110	—	—	130	145
Nickel	85	—	—	120	145
Cadmium	95	—	—	130	140

However, in numerous investigations of polymolecular films there are no confirmations that the properties of the films are analogous to those under actual conditions of their formation on friction surfaces.

Levine and Zisman [77, 78] used the measurement of coefficient of friction and investigated the capacity of a film to sustain multiple slipping for clarification of the nature of the adsorbed monomolecular layer — solid, plastic, or liquid. These measurements, besides establishing the adhesion of polar molecules to the surface, revealed the influence of the organic structure of molecules on intermolecular forces of cohesion.

Thus a bond is established between the organic structure of the molecules adsorbed by the surface and the mechanical properties of the monomolecular layer. Such a bond was established by the works of A. S. Akhmatov and his school [33, 34, 79].

As Hardy noted earlier [59], an external surface of adsorbed films which is enriched by smaller radicals turned out to be plane, where a shift is carried out most readily. Mixed absorbed films which are formed under specific conditions also contain, in addition to polar molecules, molecules of the oil base. These mixed films, however, are always unstable and are formed only in those cases when the molecules of the oil basis and polar additives are able to be completely oriented.

The adsorptive properties of boundary lubricating layers can also be connected with a change in the qualities of the microsurface.

B. V. Deryagin and Ye. F. Pichugin [80] appraised oiliness with the help of a profilometer which measured the microroughnesses of clean and lubricated surfaces. Here the index of oiliness was defined as the value, equal to 100.0 Δ,

where $$\Delta = h_{\text{ск}} - h'_{\text{ск}}$$

($h_{\text{ск}}$ – mean square deviation of height unevenness of lubricated surface, $h'_{\text{ск}}$ – the same for an unlubricated surface).

A diagram of the change of roughness a surface is shown in Fig. 2.

According to the theory of P. A. Rebinder [35], the mechanism of boundary friction is conditioned basically by an adsorptive lowering of hardness of the metals. (The specific work of dispersion is proportional to hardness).

Fig. 2. Change in roughness of a lubricated surface.

Adsorption of surface-active layers by clean surfaces of dislocations explains the further easing of the process of deformation of plasticized metal in a thin layer, which in particular is one of the essential elements of the mechanism of boundary lubrication.

A significant influence is exerted on the processes of friction and wear by oxidation of the surface and properties of the oxidized films. It was determined that during friction on the surface of a metal an oxidized layer is formed which is 10-15 Å thick.

The first step in the oxidation of metal is connected with the formation on the metallic surface of an oxide at the expense of chemisorption of atoms of oxygen by the metallic surface. The second step of oxidation is accompanied by formation of the crystal phase of oxides. For example, the following are formed: oxides of iron — $\alpha - Fe_2O_3$ (hematite), $\gamma - Fe_2O_3$ (spinel), FeO (wustite), Fe_3O_4 (magnetite); oxides of copper — Cu_2O (cuprite), CuO (tenorite), and others.

During active oxidation of metal its wear at the expense of oxidation and separation of the film of oxide can displace fatigue wear. By itself even the presence of a durable lubricating layer cannot eliminate fatigue wear, since in the presence of a film of lubrication fatigue breakdown will take place all the same inasmuch as the lubricants transmit the load onto the surface. Experiments show that the formation of iron oxide Fe_2O_3 protects the surface from damage. It is known, for example, that $\alpha - F_2O_3$ is a lubricant; it has also been established that Fe_3O_4 is an effective lubricating substance. Besides the effects of oxidation of the surface and the imparting of lubricating properties to the surface layers, the oxidizing processes taking place in the lubricant itself also essentially influences a change of its lubricating properties. This change is mainly in the direction of their improvement.

Mechanism of interaction of polar-active components of the lubricant with the metal. The combining of organic acids with a metal occurs in two stages: under the impact of dipole mirror symmetric forces, and then chemical interaction takes place with atoms from the surface of the adsorbing metal. At ordinary temperatures the adsorbed monomolecular layer (for example, stearic acid on a copper surface) includes only 25% of molecules which are bound chemically with the surface [81].

There are investigation [82, 83] which have established that the interaction of fatty acids with many metals, leading to the formation of salts, occurs only after the preliminary formation of a metal oxide.

It has also been established [84] that for the formation of a metallic salt the presence of traces of water is required.

During the so-called "clean" friction, when the interaction includes juvenile surfaces of metals, there is the possibility of an interaction of their surfaces which do not have oxides with fatty acids at ordinary temperatures with the formation of soap films [85-87].

Regarding temperatures which are characteristic for a change of disruptions of frictional bond on microcontact, then, as was shown above (Table 5), for chemical components of the type of soaps, formed on the majority of types of metallic surfaces, they have higher values than for analogous fatty acids which are physically adsorbed on a surface.

Other things being equal, i.e., with a given polar activity of molecules forming the monomolecular layer which is adsorbed by the surface, the greatest effect during friction of two materials is attained only in case when they are insoluble in each other and do not form solid alloys.

Other investigations [88-93] showed that most advantageous combination of antifriction pairs, ensuring minimum jamming and burrs during operation in bearings without lubrication, corresponds to the greatest effectiveness in the case of their lubrication by adsorbed films of fatty acids.

Lancaster [94] established that the solid films formed during boundary lubrication can under certain conditions promote the transition from boundary lubrication to hydrodynamic.

In spite of considerable successes, achieved in the direction of development of a generalized theory for calculation of wear resistance during external friction, in connection with the fact there is still not sufficiently complete data on the physical, chemical, and mechanical

properties of metals, it is doubtful whether it is possible to apply any universal analytic dependences. However, by analogy with method of appraisal of dependences, connecting the stresses and number cycles before the beginning of fatigue breakdowns, it may be possible to establish a canonical formula, stemming from the fact that pressure is one of the most important factors in processes of friction.

Such a formula can be the following:

$$p_2 = p_1 \left(\frac{N_1}{N_2} \right)^{1/k},$$

where p_1 and p_2 – specific pressures, N_1 and N_2 – their corresponding values of number of cycles during friction, k – constant; $1 < k < 2$ (we, for example, established that for a number of internal-combustion engines $k = 2$ is correct).

Hence certain objective indices are determined for appraising the effectiveness of alloying of a lubricant for internal-combustion engines based on their antiwear effectiveness [95].

With an Increase of Operating Temperature

A lubricating layer and its changing thermophysical, thermochemical, and thermomechanical properties exert a considerable influence primarily on the mechanical characteristics of a material. It is known for example [96] that

$$M_2 = M_1 \exp \alpha (t_1 - t_2),$$

where M_2 – level of mechanical properties of material at a temperature t_2; M_1 – level of mechanical properties of the same material at a temperature of t_1; α – temperature coefficient.

In a specific way, by exerting an influence on the qualitative and quantitative characteristics of the processes of generation and dissipation of heat when using additives in lubricating layer, we have the possibility to affect a change on maximum states of links and kinematic pairs of mechanisms.

The analysis given below shows how significant the characteristics of the lubricating layer (changing with a various level of alloying) can be for ensuring the required longevity of machine parts based on this particular form of their maximum state.

Let us designate:

v – rate of relative slip of elements of kinematic pairs,

t – time,

μ – coefficient of sliding friction,

p – normal load.

Then the work of forces of friction N

$$N = \mu v t p.$$

The heat Q_T generated in the process of friction will be determined by the expression:

$$Q_T = \frac{\alpha}{A} p \mu v t = 427 \alpha p \mu v t,$$

here α – coefficient, considering the share of work dissipated in heat, $A = \frac{1}{427}$ – mechanical equivalent of work. This heat is distributed between contacting surfaces and the lubricating oil

$$Q_T = Q_n + Q_o. \qquad (3)$$

where Q_H – heat, absorbed by lubricating film, Q_0 – heat, removed from the zone of friction and absorbed by the volume of oil, and also by contacting surfaces.

$$Q_0 = \vartheta \Delta t F t + \lambda F t \operatorname{grad} t,$$

where ϑ – coefficient of heat transition,

$$\Delta t = t_n - t_M,$$

t_n – temperature of oil film, t_M – temperature of oil in volume, F – area of contact, λ – coefficient of thermal conductivity of contacting surfaces, grad t – temperature gradient in surface layer

$$Q_H = cM \Delta t = chF \rho \Delta t,$$

c – heat capacity, h – thickness of film, ρ – density.

From formula (3) it is possible to obtain:

$$ch\rho F \Delta t = \frac{a}{A} p \mu v t - \vartheta \Delta t F t - \lambda F t \operatorname{grad} t,$$

dividing both sides of the equality by Ft, we have

$$\frac{ch\rho\Delta t}{t} = \frac{a}{A}\frac{p\mu v}{F} - \vartheta\Delta t - \lambda t \operatorname{grad} t,$$
$$\frac{ch\rho\Delta t}{t} + \vartheta\Delta t + \lambda t \operatorname{grad} t = \frac{a}{A}\frac{p\mu v}{F},$$

or

$$p\mu v = \frac{FA}{a}\left(\frac{ch\rho\Delta t}{t} + \vartheta\Delta t + \lambda t \operatorname{grad} t\right). \qquad (4)$$

On the basis of numerous experimental data it is clear that for a lubricating layer, having a alloying agent in its composition, it is possible to establish a definite relationship between rate and power

factors, frictional characteristics, and the index depending on the maximum carrying capacity of the lubricating layer.

In general form this dependence can be presented in the following way

$$pv\mu R = \text{const}, \tag{5}$$

where p — load, v — rate, μ — coefficient of friction, R — parameter, characterizing the extreme load capacity of the lubricating layer.

Solving jointly equations (4) and (5), we obtain:

$$\frac{ch\rho\Delta t}{t} + \theta\Delta t + \lambda t \operatorname{grad} t = \frac{\text{const}\cdot a}{R\cdot FA},$$

or

$$R\left(\frac{ch\rho\Delta t}{t} + \theta\Delta t + \lambda t \operatorname{grad} t\right) = \text{const}. \tag{6}$$

Thus, with an increase of the parameter characterizing the extreme load capacity of the lubricating layer it turns out to be possible to lower the value of the components of the polynomial which is found in parentheses, each of which in a specific way is combined with maximum temperature conditions in the zone of contact. For example, an increase of parameter R is connected with a lowering of drop of temperatures Δt, gradient of temperatures in the surface layer grad t, and others.

2. Determining the Level of Alloying of a Lubricant for Ensuring an Assigned Period of Service

In engineering practice the initial data for selection of lubricating materials for the purpose of ensuring the reliable operation of machines and mechanisms are usually limited by the extent of

viscosity of the oil. Up to now both in Soviet and also in foreign literature it is recommended to carry out the selection of lubricating materials based exclusively on this index, characterizing, as is known, only limits of preservation of hydrodynamic conditions of lubrication.

In examining of the basic forms of influences, which are characteristic for the majority of machines and which cause the gradual or sudden loss of efficiency based on achievement of maximum states, it was established that in all cases by alloying of a lubricant it is possible to affect a change of efficiency. Consequently it is also possible to pose the question about determination of the necessary level of alloying of a lubricant in order to ensure assigned longevity. For the first time such an attempt was made for a special case (relative to internal-combustion engines) [97, 98]. The contemporary state of theoretical and experimental investigations in the field of alloying of lubricating oils, and also the known methods of theory of longevity and reliability of machines [99], make it possible to approach the solution of such a problem more broadly.

For this it is necessary first of all to classify machine parts by criteria of their working assignment and maximum states of loss of efficiency when using lubricating materials alloyed in accordance with the conditions of their application.

A statistical analysis of operational data or calculation of factors determining the kinematic, dynamic, and temperature conditions of operation of subassemblies of mechanisms and machines can serve as the initial data for constructing such a classification in each specific case. For a further calculation of efficiency based on maximum states it is necessary to originate from the following objective data:

form and nature of power or kinematic factor, which during a multiple influence is the main assumed case of gradual loss of efficiency of a given component;

analytic relationships between constructive, operational, and variable (depending on assignment of alloying agent) indices of efficiency of the component.

In a general form the characteristics of loss of efficiency by a component can be presented in the following way:

$$p = \frac{kC_s}{(nh)^{1/s}}, \tag{7}$$

where p – power factor, k – coefficient, depending on conditions of exploitation, C_s – variable parameter, depending on level of alloying of lubricant, n – number of operational cycles in unit of time, h – period of service. Such initial relationships are just only for constant conditions of exploitation.

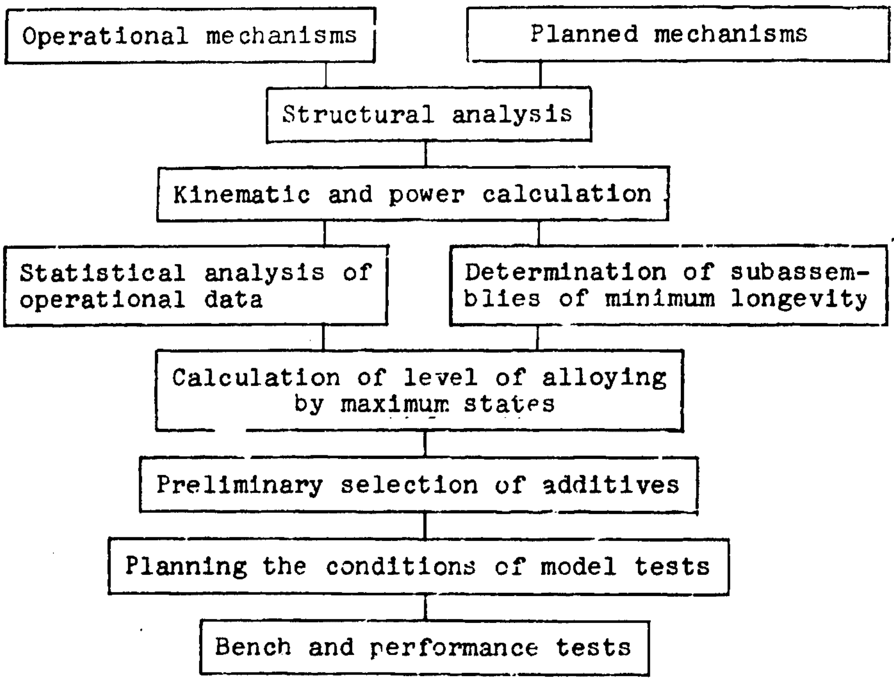
Inasmuch as real conditions most frequently are characterized by variable conditions, when in general

$$p = f_1(t)$$

and

$$v = f_2(t),$$

it is necessary to establish a certain conditional calculation constant conditions, equivalent in its influence on maximum efficiency to actual variable conditions. In operational practice most frequently it is necessary to deal with random factors which are independent from each other, which leads to the necessity of using methods of mathematical statistics [100-104]. Regardless of whether or not the initial data are materials of objective observations or whether they originate from calculation-analytic relationships, the system of consecutive approach to recommendation of the necessary level of alloying of a lubricating material for specific conditions of operation of a mechanism can be presented in the following form:



Detecting the values of variable factors, determined by the level of alloying of the lubricating material and influencing an increase of longevity of a machine relative to some specific value can in certain cases be reduced to a simple comparative investigation. Then one can compare directly the variable parameters, the values of which depend on the composition of alloying components in the lubricant.

Most frequently the most effective is such an approach to resolution of problem when the directions of alloying are determined by a generalized analysis, and the level of alloying in the selected directions – by comparative investigations. Finally, in the concluding stage of determination of the level of alloying it is necessary to appraise economic effectiveness of the selected variant of alloying.

As basis for determination of an economically justified limit of rational use of a machine for these purposes one can accept the formula proposed by V. N. Treyer [105]:

$$H_{\kappa}=\frac{\lg \mu\left(\frac{m_{n}}{m_{0}}\right)}{\lg\left(1+\frac{\kappa}{100}\right)}, \qquad (8)$$

where H_{κ} – period of service, years; k – average annual increase of productivity; m_{o} – technical norm of productivity; m_{n} – expected norm of productivity; μ = 1.5-1.6 – constant coefficient.

CHAPTER III

CLASSIFICATION OF MECHANISMS USED IN INDUSTRIAL EQUIPMENT

The accepted methodology for determining the level of alloying of industrial oils in accordance with specific conditions of their application in mechanisms of machines anticipates the necessity of a kinematic and power analysis of mechanisms, which should be preceded by a structural analysis of them.

M. V. Semenov [106] in his proposed classification of mechanisms originates not from the theory of their structure, which is reduced to the division of mechanisms into classes, orders, and families, as this is done in many cases, but from the peculiarities of the kinematic layout of the given mechanism, which reflects not only structure, but also kinematic and dynamic properties of the particular mechanism.

Thus mechanisms are divided into five basic forms: hinged, cam, mechanisms with meshing pairs, frictional, and mechanisms with a flexible connection.

Hinged Mechanisms

Mechanisms of this type are used widely in various machines and are broken down into two-link, four-link, and other hinged mechanisms.

The simplest representative of a <u>two-link mechanism</u> is the rotary kinematic pair, which finds the widest application in engines and machines of the rotary type. For example, numerous types of hydraulic, steam, and gas turbines constitute two-link flat hinged mechanisms. These include jet engines, centrifuges, centrifugal compressors, turboblowers and fans, and also generators, electric motors, and so forth. In machines of such type the mobile link (rotor) will form with a fixed link (stator) a rotary kinematic pair.

Two-link flat hinged mechanisms also include servomechanisms of "pivoting blade" type which are used widely in various constructions of hydraulic drives, where it is necessary to ensure return-rotation, different constructions of free-movement clutches, ratchets, and others.

Differences in the kinematics of flat two-link hinged mechanisms pertain basically to the degree of high speed of the mechanism.

Considerable specific importance in contemporary mechanisms belongs to rotary kinematic pairs with rolling bodies (antifriction bearings).

<u>Four-link hinged mechanisms</u> (flat) constitute a structure consisting of a two-link flat hinged mechanism and one two-guide group.

The classification of the extensive product-list of this, probably the most diverse, group of mechanisms is based on the number of rotary pairs included in the composition of four-link mechanism. It is possible, for example, to obtain three types of four-link mechanisms — four-hinged, four-link with one forward pair, and four-link with two forward pairs.

Four-hinged mechanisms include those consisting of four sections joined only by rotary kinematic pairs. Depending on the ratio of lengths of sections (cranks rockets, and connecting rods) it is

possible to obtain mono, two-crank, and two-rocket four-hinged mechanisms. Kinematic relationships, effective for finding relative shifts under assigned conditions of movement of the driving link, also are determined depending on the relationship of lengths of sections of this mechanism.

Inasmuch, as was noted above, that kinematic pairs constitute only rotary pairs, then the calculation systems used for appraisal of work conditions of the lubricant in them are analogous.

Four-link mechanisms with one forward pair are widely known in technology in the following basic modifications: crankshifts, mechanisms with a rocking link, mechanisms with a revolving link, and rocker-connecting rod mechanisms. As is obvious, two links in a four-link mechanism with one forward pair have finite dimensions, and two - infinite.

The most widely represented in various areas of technology are the crankshafts which are used in reciprocating engines. Mechanisms with a rocking link pertain to crank-rocker mechanisms. A four-link mechanism with a revolving link can be equivalent to a two-crank four-hinged mechanism (various types of engines of the rotary type). An example of a simplified mechanism with a revolving link is the vane-type pump or the kinematically analogous vane-type motor which is used widely in hydraulic transmissions.

Of the three-dimensional hinged mechanisms universal joints and mechanisms with a swash plate are widespread. The latter serve as the basis for numerous constructions of hydraulic motors and hydraulic pumps, also used in volume hydraulic transmissions.

Cam Mechanisms

The most widespread are flat cam mechanisms, the basic modifications of which we will also examine.

Based on type of mechanism there are: central cam mechanisms with forward moving follower; displaced (off-axis) cam mechanisms with a forward moving follower; cam mechanism with a rocker. In all these cases the cam is connected with a fixed link (strut) of a rotary kinematic pair. However, the joining of a cam with a strut can be carried out with a forward kinematic pair.

Especially frequently forms of geometric element of the follower or rocker are encountered which are a cylindrical surface, which is used in most cases with a passive link (cylindrical roller). This type of cam mechanisms, and also those with flat followers and with power closing carried out with help of springs, are used widely in internal-combustion engines for gas-distributing systems.

For cam mechanisms of the closed type which are working under severe conditions, the most important factor for ensuring their longevity can be the optimum selection of alloying components of the lubricant.

Maximum states of elements of kinematic pairs for closed heavy-loaded cam mechanisms which are working in lubrication are usually characterized by phenomena of contact fatigue (in other types of cam mechanisms longevity determines the wear of the profile).

Based on the method of calculation and designing of cam mechanisms used by V. A. Yudin [107] the basic dimensions of cam mechanisms are determined proceeding from assigned longevity and with a given value of contact stresses. Inasmuch as between the value of maximum normal stresses, having an effect on contact, and the number of cycles of change of this stress up to breakdown due to contact fatigue a definite dependence exists. The increase of number cycles up to breakdown will be equivalent to an increase of permissible value of maximum contact stress.

The curve of contact fatigue of a cam surface can be presented in the form of an exponential dependence [108] in the form

$$PN^{1/t} = P_0, \qquad (9)$$

where N — number of cycles of load, P_0 — value of stresses, corresponding to strength with a single load, P — effective magnitude of normal stresses, t - exponent of curve of contact fatigue.

According to Ye. I. Vorob'yev [108], the exponent of curve of contact fatigue for cams (made from St 45 with R_c = 54-56 and tempered by high-frequency current), outlined along the arcs of a circumference during work in contact with steel rollers of the same hardness, turns out to be equal to 4.1.

Having this index, it is possible to switch from actual value of maximum number of cycles of load, obtained by using various alloying components for the lubricant, to the value of contact stresses, serving as initial during the construction of cam mechanisms. With an increase of relative slipping of roller and cam there will be a significant change in the distribution and value of stresses due to the action of tangential forces [109].

Mechanisms with Meshing Pairs

Depending on the trajectories of relative motion mechanisms of this type are divided into plane and three-dimensional. The first include flat cylindrical toothed pairs with external or internal engagement and direct, slanting, or helical teeth; and the second — conical, screw, and work pairs. A general criterion for all modifications of mechanisms with meshing pairs is the presence of elements of rolling with slipping in the relative motion of the geometric elements of their links.

The classification of gear drives based on form of axoid surface [Translator's Note: aksoid – word not identified] of gears in the drive, i.e., based on form of axoid surfaces of relative motion of cogwheels, can be presented in the following way:

1) cylindrical gear drives – drive, the axoids of wheels of which are round cylinders, i.e., drive with parallel axes of wheels;

2) conical gear drives – drive, the axoids of wheels of which are circular cones, i.e., drives with transverse axes of wheels;

3) hyperboloid gear drives – drives, the axoids of wheels of which are hyperboloids of one sheet of rotation, i.e. drives with intersecting axes of wheels.

Cylindrical drives can be composed of involute or cycloid gears, Based on form of gears which are included, for example, in cylindrical straight cogs we can classify them as:

straight-toothed cylindrical gears, for which the angle of inclination of tooth line is equal to zero;

slanting-toothed cylindrical gears, for which the angle of inclination of tooth line is not equal to zero and is constant;

curved-toothed cylindrical gears, for which the angle of inclination of tooth line is variable according to the length of the tooth.

Conical gear drives can be composed of conical gears of the following forms:

straight-toothed with an angle of inclination of tooth line is equal to zero;

slanting-toothed, the tooth line of which on unrolling of monotype coaxial surface constitutes a straight line, tangent to the circumference, the center of which coincides with the center of unrolling;

curved-toothed, the tooth line of which on unrolling of the monotypic coaxial surface constitutes a curve.

In particular, curved-toothed conical gears, the tooth line of which on unrolling of a monotypic coaxial surface are arcs of the circumference or involute (its center coincides with the center of unrolling will be accordingly:

conical gears with a circular tooth line (spiral-toothed wheel);

conical gears with an evolvent tooth line (palloid gears or Klingel'berg wheels).

Hyperboloid transmissions can be:

screw (helicoidal), if they are composed of involute cylindrical gears;

hypoid – are composed of conical cogwheels.

Hyperboloid also include worm gears which are hyperboloid gear drives with linear contact, where the producing surface of one of the links of this drive coincides with the theoretical working surface of the teeth of the other. Worm gear drives can be cylindrical or toroid (globoid). In the first case the gear includes a convolute worm, in the second – a toroid.

A convolute worm is a cylindrical worm, the theoretical working surface of which is formed by the movement of the straight line forming, which is tangent to a certain coaxial cylinder (initial) in points of the helix on it and forming a constant angle with the tangent to the helix.

In a convolute worm the theoretical working profile of a tooth in a plane tangent to the initial cylinder is a straight line, and in the face section – a shortened or extended evolvent. In a case when the radius of the initial cylinder is equal to zero, this will be an Archimedian work (in an Archimedean work the theoretical working profile of the tooth in the face section is an Archimedean spiral). When in a convolute worm the generatrix is tangent to the helix on the initial (basic) cylinder, this will be an involute worm (in an involute worm the theoretical working profile or the tooth in the face section is involute).

A toroid (or globoid) worm is a worm, the theoretical working surfaces of the teeth of which are formed by the uniform motion of a straight line roll along a circumference (profile), located in the axial plane of the worm, with the simultaneous rotation of this plane around the axis of the worm.

Finally, the last two groups of mechanisms – frictional and mechanisms with a flexible connection – are of lesser interest for characterizing the conditions of application of lubricating materials. The application of lubrication in these forms of mechanisms is limited to a small number of designations, inasmuch as here the most important conditions of closing of kinematic pairs include high coefficients of friction, which excludes the application of lubrication.

C H A P T E R IV

METHODS FOR APPRAISAL OF THE INFLUENCE OF ADDITIVES ON A CHANGE OF CERTAIN LIMITING STATES OF MACHINE COMPONENTS AND SUBASSEMBLIES

The efficiency of mechanisms is determined basically by their resistance to different forms of breakdowns, characterized by limiting states, the manifestation of which usually is connected with the quality of lubricating oils and especially of additives which are used.

The basic forms of maximum states include: wear of friction surfaces of kinematic pairs; burrs, jamming, and gripping of surfaces; breakdowns connected with contact fatigue of material.

Each of these forms of breakdowns depending on the predominant factors of mechanical, chemical, or thermal origin is characterized by peculiarities which are inherent to conditions of work of the particular kinematic pair. They may also be in a complex interdependence, leading in the end to any one leading form of breakdown. In the practice of research work on the preliminary appraisal of operational qualities of oils with additives it is important to reproduce on the testing units conditions which are as close as possible to actual.

In the development of methods for appraisal of the influence of additives on a change of limiting states the following basic properties are noted: antiwear, antiburr, antipitting.

1. Appraisal of Antiwear Properties of Oils on a Four-Ball Friction Machine

In the appraisal of operational qualities of oils for contemporary machines and mechanisms the most important factor is the capacity of oils to prevent intensive wear of friction surfaces, i.e., to increase the longevity and reliability of machines.

Preliminary elimination tests of the qualities of oils and oils with additives should make it possible to compare their antiwear properties. Years of operational experience at the IKhP of the Azerbaydzhan SSR Academy of Sciences and several other research organizations on the appraisal of antiwear qualities of oils and oils with additives showed the considerable advantages of four-ball friction machines over other types of machines. Therefore as the base for carrying out of this work we took the four-ball friction machine (ChShM).

Over a number of years considerable material has accumulated in the area of using this machine for appraisal of the quality of lubricating oils. At the basis of almost all the methods of appraisal of antiwear properties of oils and oils with additives on the ChShM lie prolonged tests with various load and speed conditions and the application as the appraisal parameter of the value of linear or weight wear of balls of different diameter. For example, a method is used which provides for carrying out of tests for 2 hours at 1180 r/min and small loads (7, 12, and 20 kg) on ball with a diameter of 12.7 or 25.4 mm. The curve of dependence of diameter of friction track on load (Fig. 3) characterizes the antiwear properties of the particular oil. The results obtained for oils which are sharply different in quality are found in rather small limits of values of diameter of friction track – from 0.2 to 0.8 mm.

R. M. Matveyevskiy [110] used the method of appraisal of antiwear properties of various oils under the following conditions: diameter of balls 12.7 mm, rate of slip 0.56 m/s, duration of test 8 hours. Appraisal parameters were volumetric wear of balls, value

of friction coefficient, and temperature rise during the time of the tests. Characteristic curves, obtained for two different oils, are shown in Fig. 4.

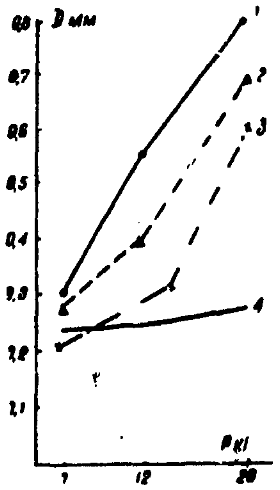


Fig. 3. Dependence of wear on load. Oils: 1 – MS-20; 2 – GOST 4003-53; 3 – $TA_п$ GOST 8412-57; 4 – MS-20 + 2% additive.

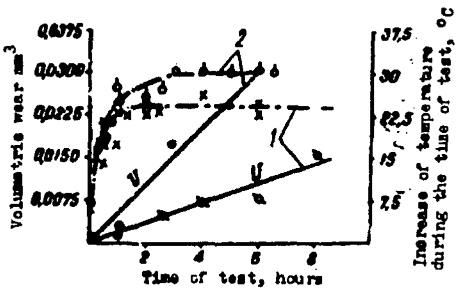


Fig. 4. Dependence of volumetric wear and temperature of oil on the time of the test (solid lines – wear, dotted – increase of temperature) 1 – SU machine oil ($P_к$ = = 16.75 kg), 2 – turbine Л ($P_к$ = 16.75 kg).

As can been seen, for SU machine oil after 8 hours of testing wear comprised 0.015 mm^3, and for turbine Л with 6 hour test duration volumetric wear reached 0.03 mm^3. These values, being insignificant by themselves, undoubtedly will have still lower values in a case when tests are conducted with antiwear or other additives. The latter limits the use of the method for differentation of additives which differ little in antiwear effectiveness and which ensure a sufficiently low level of wear under normal conditions.

Curves of dependence of wear on load for appraisal of the qualities of oils were used by Van-Dik and Block [111], using short-term and prolonged test conditions. In the first case for each load the duration of the test comprised 1 min, in the second – 8 hours. The tests were conducted on balls with a diameter of 12.7 mm at 1500 r/min for the central ball. This corresponded to a relative speed of 0.68 m/s. The first method differs little from the usual method of one-minute tests, and the second makes it possible to obtain a noticeable differentation of quality only with considerable expenditures of time. This is illustrated graphically by the nature of the curves obtained on the graph (Fig. 5). The series of additives tested by this method differs little from each other and from oil without an additive.

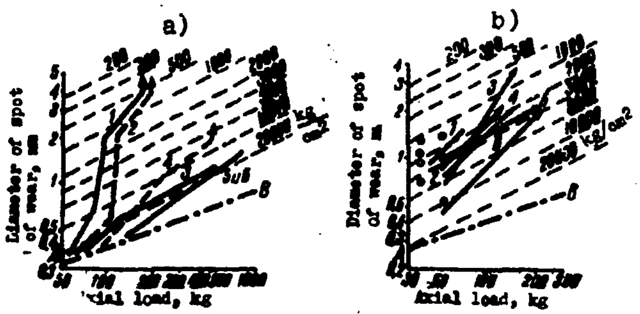


Fig. 5. Dependence of wear on load during short-term (a) and prolonged (b) tests. AB – line of diameter of track of elastic deformation, 1 – oil without additive, 2 – oil with additive of fatty acid, 3 – oil with sulfur additive, 4 – oil with sulfur- and chlorine-containing additive, 5-6 – oil with phosphorus additive.

The well-known MHL [Mean Hertz Load] method [112], corresponding to GOST 9490-60, characterises oils by two indices, one of which determines the maximum efficiency of the oil, i.e., a certain conditional limit, above which the friction surfaces which are lubricated by the test oil are noneffective and the second – intensity of wear of the friction surfaces, occurring up to the moment when they are recognized as noneffective.

As an index of maximum carrying capacity (efficiency) of oil it is possible to select one of two characteristic loads: directly preceding a sharp increase in the diameter of the spot of wear on the balls due to their jamming ($P_{к}$) or the load at which fusing of the balls occurs (fusing load P_{c}).

Selection of fusing load as the index of maximum efficiency of an oil instead of the widely used jamming load is explained by the assumption that index P_{c} possesses a series of advantages over $P_{к}$. In particular, for highly effective additives to oils a clearly expressed fracture of curves of dependence of wear on load, corresponding to maximum jamming load $P_{к}$, index P_{c} characterizes the complete loss of efficiency of oil, whereas oil beyond the extremes of $P_{к}$ still preserves a certain efficiency.

However, in our opinion these considerations do not determine with sufficient objectiveness the validity of such a preference. The fact is that complete loss of efficiency of a lubricant which is usually used under actual conditions of work in heavily loaded gears is not observed. In this meaning the index P_{c} characterizes certain properties of oils which are specific for conditions lying beyond the borders of those actually observed. Furthermore, experience in the appraisal of antiburr properties of oils with additives shows that in most cases index P_{c} does little to supplement the appraisal of quality of oil based on the generalized index of wear, and its value for widely used oils actually changes in very insignificant limits.

Index $P_{к}$ has a distinctly expressed meaning, characterizing such a most important functional property of a lubricating layer as its maximum efficiency up to a certain qualitative jump, determining the transition from one condition of lubrication to another.

The absence of a break on curve $d = f(P)$ already by itself is a sufficiently objective index of quality of the lubricating layer and indicates preservation by this layer of stable antifriction, antiburr, or some other functional characteristic in a wide range of pressures and temperatures.

The generalized index of wear itself, which is one of the appraisal criteria of oils according to the method regulated by GOST 9490-60, is analogous to the MHL index which is accepted in accordance with federal standards of the United States.[1] It amounts to a comparative appraisal of the level of wearability of friction surfaces in a wide range of loads, preceding the fusing load.

With such a generalized appraisal of level of wear according to the results of a series of experiments on various loads, each of which has a duration of 10 s, only the diameter of the friction track is taken into account regardless of whether wear takes place under comparatively normal conditions, without the phenomena of jamming, or it is obtained due to an intense burr, usually preceding fusing.

In summing up the results of a series of experiments up to a certain prefusing load, defined as the load of the last step before P_c, an average quantitative characteristic of wear is obtained, the physical meaning of which can be expressed by the equation

$$\mathrm{OPI} = \frac{\sum\limits_{i=1}^{i=m}\left(\frac{P_0}{d_и/d_н}\right)}{m}, \qquad (10)$$

where P_0 – axial load, $d_и$ – diameter of spot of wear at load P_0, $d_н$ – diameter of track of elastic deformation, calculated according to the Hertz theory and corresponding to load P_0, m – number of experiments (m = 20 is taken).

Thus, the OPI [generalized wear index] is the arithmetic mean of 20 values, each of which constitutes the ratio of axial load to relative wear, i.e., to a value, showing by how many times the diameter of the track of absolute wear is greater than the diameter of the track of elastic deformation.

[1]USA Federal Specification VV-L-791 c, Method 6503.

Such a ratio is selected as initial for determination of the quantitative characteristic of wear on the assumption that wear starts on the actual site of elastic deformation.

A somewhat more expanded characteristic of generalized index of wear is given below.

In examining the dependence $d_H = f(P_0)$, where d_H – diameter of track of wear and P_0 – axial load, we establish that in a double logarithmic system of coordinates a broken line is obtained. Let us assume that dependence

$$d'_H = cP_0^m \tag{11}$$

represents the function of "levelled wear" d'_H from axial load P_0. Let us assume that

$$c = \frac{0,08763}{c_1},$$

then

$$d'_H = \frac{0,08763}{c_1} P_0^{m-1} P_0. \tag{12}$$

For determination of dependence of diameter of track of elastic deformation (according to Hertz) of steel balls with a diameter of 12.7 mm it holds true that

$$d_H = 0,08763\ P_0^{m-1}, \tag{13}$$

then

$$d'_H = \frac{d_H}{c_1} P_0,$$

from which

$$c_1 = \frac{P_0 d_H}{d'_H}.$$

We compare the resulting expression for c_1 with the formula for determination of the OPI. Obviously they are identical under the condition that

$$\left(\frac{P_0}{d_н} d_н'\right)_1 = \left(\frac{P_0}{d_н} d_н'\right)_2 = \ldots = \left(\frac{P_0}{d_н} d_н'\right)_i = \ldots = \left(\frac{P_0}{d_н} d_н'\right)_n .$$

i.e., under the condition that

$$\frac{P_н}{P_н} d_н = \text{const.}$$

Here it is accepted that $d_H' = d_n$. In this case constant c_1 represents the OPI value at a certain levelled wear.

$$\text{Thus, OPI} = \frac{P_0 d_н}{d_н'} \qquad (14)$$

where P_o – axial load, kg; d_H – diameter of site of elastic deformation according to Hertz, corresponding to axial load P_o, mm; d_H' – diameter of track of friction at levelled wear, corresponding to axial load P_o, mm.

From equations (13) and (14) we obtain

$$\text{OPI} = \frac{P_0 \cdot 0{,}06763\, P_0^{m-1}}{d_н'} = \frac{0{,}06763\, P_0^{m}}{d_н'}, \qquad (15)$$

$$\text{or } d_н' = \frac{0{,}06763\, P_0^{m}}{\text{OPI}}.$$

Curves of levelled wear are obtained when $m = 1.333$ is taken. In this case with an increase of OPI value the curves are displaced to the right.

From formula (14) it is possible to establish that the OPI constitutes the ratio of axial load to relative wear $\frac{d_н'}{d_н}$ with this load. Here relative wear is the ratio of the value of levelled wear d_H' to the track of elastic deformation at the particular load.

The connection between levelled and actual wear is established if instead of the accepted $\left(\frac{P_0}{d_n} d_n\right)_1 =$ const the arithmetic mean from n values is introduced in the calculation.

Also widespread are methods of appraisal of lubricating properties of oils on a four-ball friction machine based on the value of pressure on the lubricating film at the point of bend of the curve of dependence of average diameter of friction track on the lower balls on the load which is effective at the point of contact.

Force P, applied along the axis of the upper ball of the four-ball pyramid, is separated into 3 forces P_1, the line of action of which passes through the centers of the contacting balls. Thus, P_1 = 1/3 P.

We will determine angle ϕ (Fig. 6):

$$\varphi = \arcsin\frac{a}{2r}, \text{ but } \frac{r}{a} = \cos 30^\circ, \text{ i.e., } a = \frac{r}{0{,}866}.$$

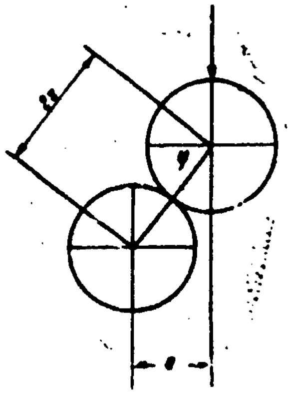


Fig. 6. Diagram showing contact of balls.

Consequently, $\varphi = \arcsin\frac{r}{0{,}866 \cdot 2r}$. From here $\varphi = 35^\circ$.

On the basis of this

$$P_1 = \frac{P:3}{\cos 35^\circ} = 0{,}407\,P \approx 0{,}41\,P.$$

In the case of elastic compression of two spherical surfaces the maximum specific pressure at the point of contact is determined by the formula of Hertz:

$$P_{max} = \sqrt[3]{\frac{1{,}5 P_1 E_2}{\pi^3 (1-\mu^2)^2}\left(\frac{d_1+d_2}{d_1 \cdot d_2}\right)^2}. \qquad (16)$$

Taking $\mu = 0.3$ and considering that $d_1 = d_2$, we obtain

$$P_{max} = \sqrt[3]{0{,}0584\,\frac{P_1 E^2}{(d/2)^2}}. \qquad (17)$$

Average pressure on the contact surface

$$P_{cp} = \frac{2}{3} P_{max}.$$

Thus under these circumstances of tests $P_c = k\sqrt[3]{P_1}$.

Furthermore, average pressure on area of contact is determined based on load and diameter of friction track:

$$\sigma = P_1 \frac{4}{\pi d^2} = 0{,}522\,\frac{P}{d^2}, \qquad (18)$$

where d — diameter of friction track.

The pressure which is conditionally apportioned to the lubricating film at loads P = 200 and 300 kilograms is also estimated.

The described method of appraisal of strength of an oil film based on $P_к$ suffers with that deficiency that oils without additives differ from each other based on this index by no more than 2 times (with additives up to 5 times), while in operation their behavior is thoroughly different. This deficiency is aggravated by the low accuracy of determination of $P_к$.

Therefore Klimov developed a method of appraisal of antiwear properties of oils based on the average rate of wear of balls in the period of jamming. The method amounts to the following. If one were to graphically depict the dependence of friction coefficient, determined by the moment of friction, on the time of rotation τ, then for loads $<P_{\text{к}}$ we obtain almost the constant value μ = 0.12 for the entire time of the test τ = 60 s (Fig. 7). If, however, an analogous test is conducted with $P > P_{\text{к}}$, then μ at first will increase to 0.38, and then anew will drop almost to the initial value (see Fig. 8); in the period τ_2 intensive wear of balls occurs. The graph shows that the latter is not the same for various oils. Thus, according to Klimov synthetic oil SS = 908 has a τ_2 of jamming of 40.5 s, AS-9.5 Binagadinskaya oil – 20.6, MS-20 + AzNII TsIATIM-1 – 4.25, and hypoid GOST 4003-43 – 60 s.

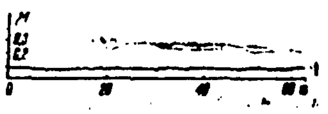


Fig. 7. Dependence of friction coefficient μ on time τ at $P < P_{\text{к}}$.

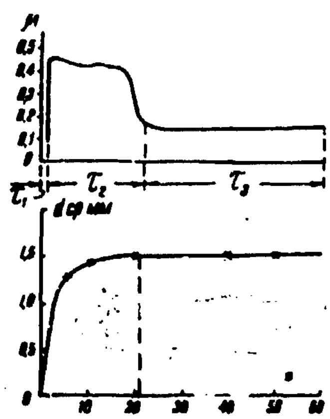


Fig. 8. Dependence of friction coefficient μ on time τ with $P > P_{\text{к}}$.

Determining by experimental means the time of jamming τ_2 and the volume of friction of the metal taken for 60 s:

$$V = 2\frac{1}{6}\pi \cdot h\,(3a^2 + h^2).$$

where $a = h(2r - h)$ – radius of base of segment, h – height of spherical segment, r – radius of ball, we obtain the average rate of jamming

$$v_{cp} = \frac{v}{\tau_s} \ \frac{\mathrm{mm}^3}{\mathrm{s}}.$$

Further Klimov showed that v_{cp} is practically a linear function of P:

$$v_{cp} = v_0 + k(P_1 - P_0),$$

where $k = \mathrm{tg}\ \alpha$ (see Fig. 9).

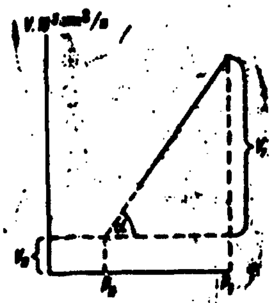


Fig. 9. Dependence of average rate of wear of balls V_{cp} on load P.

<u>Selection of rate, load, and temperature regimens for tests on a four-ball friction machine</u>. Load at the beginning of friction of four-ball pyramid P_m, as it was shown, is determined according to Hertz [113]

$$P_m = k\sqrt[3]{P_к}$$

where $P_к = 0.41\ P$ (P – axial load on pyramid), $k = 51.74$ (for balls with a diameter of 19 mm) and 67.66 (12.7 mm). Thus a change of load can be attained either by variation of the diameter of the balls or by changing the axial load on the pyramid. Inasmuch as a change in the diameter of the balls involves a change of relative rate of slip of the rotating upper ball relative to the fixed lower, we will examine the dependence of wear of balls on the rate of relative slip.

R. M. Matveyevskiy [110] obtained the dependence of wear of balls on the rate of relative slip and temperature (Fig. 10). As can be seen, up to a rate of slip of 0.8 m/s wear depends little on it.

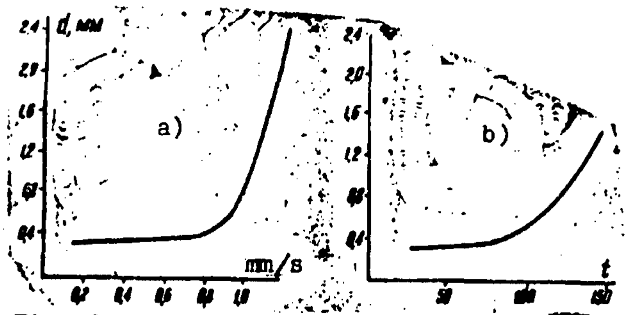


Fig. 10. Dependence of wear of balls on relative rate (a) and temperature (b).

Thus with an increase in the diameter of the balls the rates of slip have little affect on their wear. On the other hand, an increase in the diameter of the balls decreases the value of initial contact stresses, as a result of which work with their assigned values can be carried out at loads less than those corresponding to the beginning of jamming.

For a comparison of results, obtained during friction of balls with various diameters, we will examine the data shown in Fig. 11.

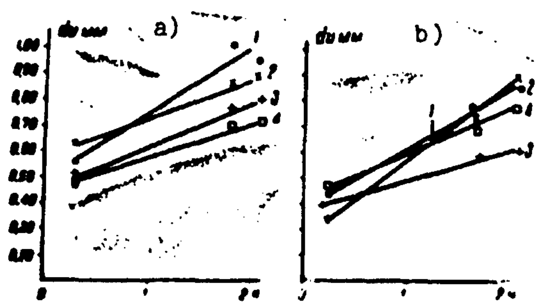


Fig. 11. Results of tests of various oils on a ChShM when using balls with a diameter of 12.7 (a) and 19 mm (b) 1 – motor oil, 2 – MS-20, 3 – nigrol, 4 – MK-22.

Tests were conducted at 1475 r/min and a load of 1.2 kg, which corresponds to P_m = 16,000 kg/cm^2 for balls with a diameter of 19 mm and 21,000 kg/cm^2 – 12.7 mm. Considering that in both cases on contact stresses take place which correspond to those under conditions of engagement of straight-toothed and spiral-conical cogwheels, it is expedient to conduct the tests under the shown conditions. Inasmuch as up to 100° the wear of balls, as can be seen in Fig. 10b, depends little on temperature the tests can be conducted at room temperature (without special preheating), as this is usually done during tests on a four-ball machine.

Aviation oils MK-22 and MS-20, nigrol, – based on GOST 542-50, and AK-15 motor oil were tested. Results of the tests are shown in the form of graphs in Fig. 12, and absolute values of wear are given in Table 6.

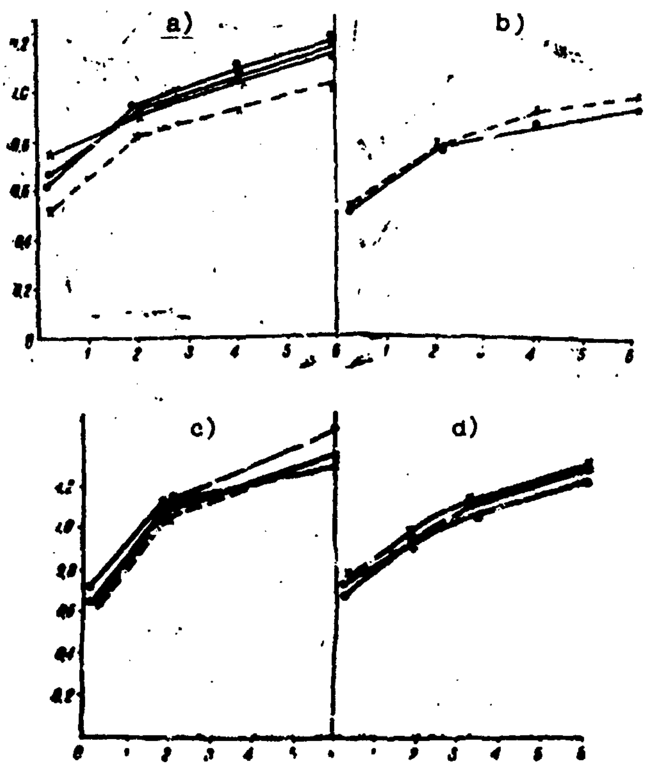


Fig. 12. Dependence of wear on the time of test on a ChShM. a) MK-22; b) nigrol; c) AK-15 (TU-8 – 61); d) MS-20.

Table 6. Wear of balls during tests of various oils, mm·10^{-2}.

Sample of oil and No. of experiment	Duration of work			
	15 min	2 h	4 h	6 h
MK-22				
1	63	95	111	123
2	66	94	158	120
3	75	92	105	117
4	52	82	92	106
Nigrol				
1	51	78	89	94
2	52	77	93	99
MS-20				
1	69	96	115	124
2	74	94	108	120
3	76	101	116	127
AK-15				
1	76	113	124	129
2	66	112	131	146
3	64	109	121	135

As will be shown below, by conversion of the data obtained into values of volume wear based on diameter of the friction track linear dependence of wear on time of work is obtained. This makes it possible to appraise the comparative intensity of wear.

Thus, it turns out to be possible to recommend the test conditions accepted above. As can be seen on the curves of wear, work for more than 6 hours increases the diameter of the friction track little, as a result of which this length of test is considered fully sufficient.

Data from the literature [110, 111, 114] testify that in most cases antiwear properties of lubricating oils according to the results of tests on four-ball friction machines are appraised by the change in the diameter of the track on the lower balls. Graphically the dependence of the diameter of a spot of wear on time is expressed by a curve of the type $y = x^n$, where $n < 1$.

The nature of this dependence turns out to be somewhat different according to Bingham and Wright [115]. According to them with an overall test duration of 80 min in the interval of time from 10 min to the end of the test n = 1 took place.

If one were to reconstruct the graph of dependence d = f(t) in coordinates v = f(t), then inasmuch as $v = 1/6\pi h(3a^2 + h^2)$, where v – volume of spherical segment with height h, having a base of radius a, the curve of dependence of volumetric wear on time of test is rectified, as this can be seen from the graphs given in Fig. 13.

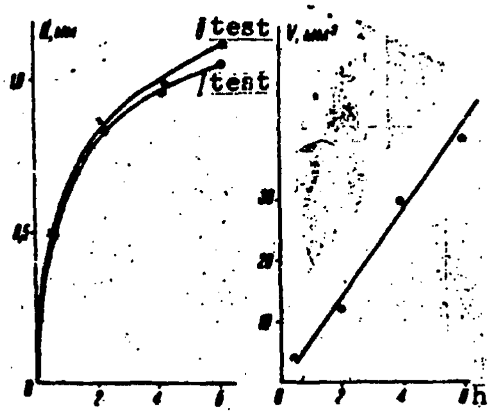


Fig. 13. Results of tests on a ChShM.

Obtaining the linear dependence of volumetric and, consequently, weight wear on time, it is possible to take as the criterion, appraising the flow of the process of wear, the index of intensity of wear

$$i = \frac{\Delta l}{\Delta t} = \frac{l_2 - l_1}{t_2 - t_1} \text{ mg/h}.$$

According to parallel tests, the results of which are given in Table 7, it is clear that the limiting value of deviation of values of wear on the balls during separate determinations comprises no more than 13.4% in the direction of larger values and 11.4% in the direction of smaller values. In Table 6 results were given from parallel determinations which were conducted for other initial oils (nigrol,

MS-20, MK-22). Sensitivity of the method was appraised by the results of tests of various additives to oils. These additives were of various concentrations. The data cited show that selected method makes it possible sufficiently clearly to separate additives and concentrations of them according to wear of balls.

Table 7. Results of parallel tests of AK-15 oils for various times of work.

No. of experiment	15 min			2 h			4 h			6 h		
	Wear, mg	Deviation		Wear, mg	Deviation		Wear, mg	Deviation		Wear, mg	Deviation	
		abs.	%		abs.	%		abs.	%		abs.	%
1	0,76	+0,09	+13,4	1,13	+0,03	+7,6	1,24	0	0	1,29	−0,07	−5,1
2	0,66	−0,01	−1,5	1,12	+0,07	+6,6	1,31	+0,07	+5,6	1,46	+0,1	+7,3
3	0,64	−0,03	−4,5	1,09	+0,04	+3,8	1,21	−0,03	−2,4	1,35	+0,01	+0,7
4	0,62	−0,05	−7,4	1,04	−0,01	−0,95	1,21	−0,03	2,4	1,35	−0,01	−0,7
5	0,62	−0,05	−7,4	0,99	−0,06	−5,7	1,20	−0,04	−3,1	1,38	+0,02	+1,5
6	0,71	+0,04	+6	0,93	−0,12	−11,4	1,26	+0,02	+1,6	1,30	−0,06	4,4
7	0,69	+0,02	+3	1,07	+0,02	+1,8	1,27	+0,03	+2,4	1,37	+0,02	+1,5
Average	0,67			1,05			1,24			1,36		

For a comparison of data obtained with the results of tests of oils with additives, the operational qualities of which are well-known tests were made of two standard additives - khloref-40 and anglamol-70. Indices obtained are given in Table 8.

The sample of oil with 2% additive of khloref-40 which gives the best results, as this is possible to conclude from results of bench tests of a series of samples of experimental transmission oils intended for main transmission with a hypoid pair on trucks (conducted in 1963 at the Likhachev Plant), possesses the best operational properties. Analogous results are also obtained during performance tests.

Table 8. Results of tests of additives to oils.

Additive	Concentration, %	Wear of balls (mm) in			
		15 min	2 h	4 h	6 h
[illegible] styrene ([illegible]-30)	0,5	0,60	0,93	1,33	1,52
	1,0	0,53	0,84	1,09	1,18
	3,0	[illegible]	0,60	1,04	1,09
	5,0	0,51	0,89	1,01	1,21
Alkylphenol, condensed with formalin	0,5	0,72	1,05	1,34	1,57
	1,0	0,62	1,21	1,92	2,00
	3,0	0,60	0,93	1,11	1,48
	5,0	0,45	0,54	0,58	0,61
	7,0	0,43	0,53	0,53	—
Products of condensation of aromatic with chloral					
Benzene	3	0,55	1,09	1,26	1,24
Isobenzene	3	0,60	1,10	1,22	1,28
Polyalkylbenzene	[illegible]	[illegible]	[illegible]	1,21	1,22
Isobutylphenol	[illegible]	[illegible]	[illegible]	[illegible]	1,18
Toluene	[illegible]	[illegible]	[illegible]	[illegible]	1,35
	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Xylene	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Khloref-40	2	[illegible]	[illegible]	[illegible]	[illegible]
Zaglenol-70	10	[illegible]	[illegible]	[illegible]	1,32

Results of performance tests of samples of transmission oils MS-20 and AK-15 were compared with the appraisal of their antiwear properties, obtained on friction four-ball machine by the described method. For calculating the intensity of wear based on curves of wear the latter across the diameter of the friction track was recalculated for volume. The results of these tests are compared in Table 9. As can be seen, AK-15 oil gives wear which is approximately 1.5 times greater.

Table 9. Comparison of results of wear tests for the ChShM with results of performance tests.

Type of test	AK-15		MS-20	
	Abs.	%	Abs.	%
On ChShM (volumetric wear, mm^3)	23,4	158	14,8	100
Operational				
Wear of gearshift cogwheels, mm	0,3	150	0,2	100
The same for end transmission	0,14	140	0,10	100
Wear of MTZ gears [MTZ - make of tractor]	0,04	100	0,04	100

Thus the method of appraisal of antiwear properties of lubricating oils and oils with additives during prolonged work on a four-ball friction machine is reduced to the following.

The machine constitutes a pyramid made of four balls and should satisfy the following requirements:

the upper ball is secured stationarily in a revolving spindle, and the three lower balls – in a fixed cup filled with the tested oil;

speed of the spindle while idling – 1410-1440 r/min;

during the tests the load is applied in such a way that axial force, acting along the axis of the spindle, presses the upper and lower balls to each other;

capacity of the cup with the three lower balls secured in it – 8 cm^3, and the level of the tested oil is brought to the tops of the balls;

hardness and vibration resistance of construction: radial play of the upper ball, measured at a distance of 4 mm from the lower point, cannot be greater than 0.02 mm; the cup with the lower balls should be self-adjusting.

For measurement of friction tracks on the balls a tool microscope is used. Before beginning tests of a new sample of oil the balls in the clip and the upper ball in the spindle are washed with gasoline and dried. For every test a new set of balls is used.

The tested balls are secured in the spindle of machine and in the lower clip, into which the oil is poured. A load of 37 kg is applied to the spindle (it corresponds to 1.42 kg on the lever of the loading device), after which an electric motor is turned on. After 15 min, 2, 4, and 6 hours of work on the three lower balls of the pyramid of the friction subassembly measurements are made of the spots of wear on each ball in two directions – lengthwise and across the channels.

According to the results of the measurements the average diameter of a spot of wear is found. It is set as the arithmetic mean of six measurements of three balls after 15 min, 2 and 6 hours of work.

Determination of average intensity of wear in the form of a conditional mean of rate of loss of weight of the balls is done in the following way. If one were to designate: g – wear of balls (10^{-3} mm^3), t – time of test (hours), k – intensity of wear (mm^3/h), g_o – initial wear (mm^3), then using the method of least squares it is possible to write:

$$\Sigma gt + k\Sigma [t]^2 - g\Sigma t = 0,$$
$$\Sigma g - k\Sigma t - g_o S = 0$$

where S – number of tests.

$$\text{Then } k = \frac{S\Sigma tg - \Sigma t \Sigma g}{S\Sigma [t]^2 - [\Sigma t]^2};$$
$$g_o = \frac{\Sigma g \Sigma [t]^2 - \Sigma t \Sigma tg}{S\Sigma [t]^2 - [\Sigma t]^2}.$$

In certain cases of tests direct measurements of the volume of a metal Δv, taken from friction surfaces, showed that with identical diameters of friction track this value turns out to be different (see, for example, the curves reflecting the dependence of value of volumetric wear on the diameter of friction track for oil without additive and with additive – Fig. 14). Consequently, in the process of wear when the oil contains additives there is a strengthening of the surface layer of the metal, which changes the relationship of deformations preceding wear at the point of contact of the balls.

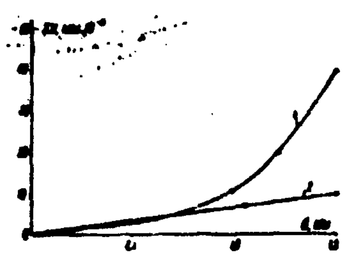


Fig. 14. Results of tests of AK-15 oil without an additive (1) and with 1% INKhP-30 (2).

For appraising the effect of strengthening of the surface layer of metal in the process of friction it is possible to use an index, characterizing the relative strength of the friction surface. Such an index can be the coefficient of strengthening

$$k = \frac{h_1 - h_2}{h_1} = 1 - \frac{h_2}{h_1},$$

where h_1 – theoretically determined value of height of spherical segment h with the complete absence of "strengthening of surface" with a rated value of diameter of friction track d; h_2 - actually obtained value h with the same d.

In an ideal case, when the diameter of the friction track corresponds to the diameter of the spot of elastic contact,

$$h_2 = 0 \text{ and } k = 1.$$

In another limiting case, when

$$h_1 = h. \qquad k = 0.$$

Below a method is given for the grapho-analytic determination of coefficient k based on experimental data obtained during the friction of balls.

Let us designate (see Fig. 15): a – radius of base of segment, $a = \frac{s}{2}$, v – volume of segment. It is known that

$$a = \sqrt{h(2R - h)},$$
$$v = \pi h^2\left(R - \frac{h}{3}\right) = \frac{1}{6}\pi h(3a^2 + h^2),$$

$$R = \frac{a^2 + h^2}{2h}.$$

From expression

$$v = \frac{1}{6}\pi h(3a^2 + h^2)$$

we obtain

$$h^3 + 3a^2h - \frac{6v}{\pi} = 0.$$

Designating $3a^2 = 3p$

$$2q = -\frac{6v}{\pi}$$

we have:

$$y^3 + 3py + 2q = 0.$$

Since the discriminant of the given equation

$$\Delta = p^3 + q^2 > 0,$$

the real root of the equation will be

$$y_1 = u + v,$$

where u and v – real values of cubic roots;

$$u = \sqrt[3]{-q + \sqrt{\Delta}},$$
$$v = \sqrt[3]{-q - \sqrt{\Delta}}.$$

Thus,

$$h = \sqrt[3]{-q + \sqrt{\Delta}} + \sqrt[3]{-q - \sqrt{\Delta}}. \qquad (19)$$

As is known, the abscissas of points of intersection of curve $y = x^3$ and straight line $y = -(px + q)$ give the solution of the equation

$$x^3 + px + q = 0.$$

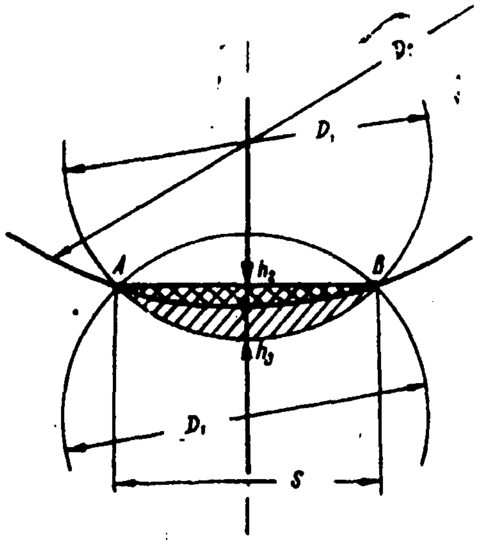


Fig. 15. Diagram of wear of a ball.

Thus, having constructed the dependence $y = x^3$, according to values of coefficients p and q we construct the straight line

$$y = -(px + q).$$

For our conditions $p = 3a^2$,

$$q = -\frac{6v}{x},$$

where a – radius of circumference of spot of wear (mm), v – volume of metal, taken from the friction surface.

Thus having determined the value $x = h_2$, analogously it is possible to find h_1 by solving the equation

$$h_1^2 + a^2 - 2Rh_1 = 0 \qquad (20)$$

by the method described above.

Tests were made of various concentrations of OS (INKhP-30) additive to AK-15 oil and the following data obtained:

Conc. of additive, %	d of friction track, mm	h_1, mm	h_2, mm	k
0.5	1,52	0,26	0.09	0,65
1	1,18	0,17	0.04	0,76
3	1,09	0,14	0,92	0,79
5	1,21	0,16	0 04	0,71

The data given show that the optimum concentration of OS additive, proceeding from its influence on the effect of "strengthening" of the surface layer, is 3%. This same concentration corresponds to the least susceptibility to wear, determined by the change in the diameter of friction track.

Thus as appraisal indices it is possible to select the actual wear of balls based on diameter of friction track and the coefficient of strengthening of the surface.

2. Method of Testing the Lubricating Capacity of Oils with Additives on an MI Machine

In Fig. 16 a general view (a) of the machine and a cross section of the load mechanism (b) are shown.

The tested friction pairs, constituting the outer races of radial thrust bearings 7204, are secured accordingly on the upper and lower spindles.

Rotation is transmitted from a power-driven electric motor to the drive shaft and further through a pair of cylindrical gears to the spindle. The position of the gear on the shaft is fixed with a screw. In the case of the fixed upper sample of gear it is displaced to the right from the stop and is fixed with a pin.

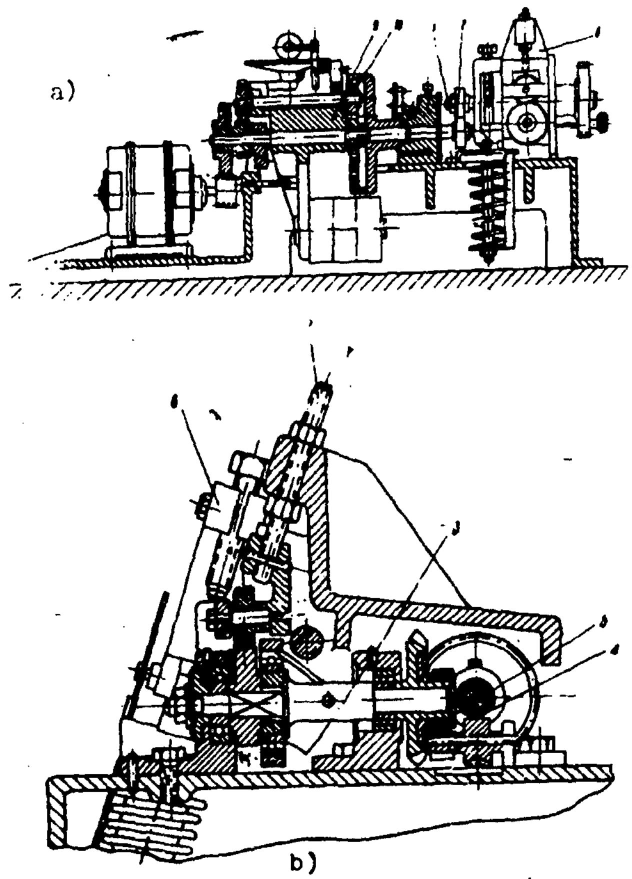


Fig. 16. MI-8 friction machine. 1 – upper spindle; 2 – lower spindle; 3 – transverse shaft; 4 – worm gear; 5 – cogwheel; 6 – carriage, 7 – adjusting screw; 8 – clasp; 9 – body of pendulum; 10 – cogwheel.

Load on the tested samples is carried out by a spring, the force of which is transmitted with help of a clasp, joined to the shaft by a pin. Load is changed by a tension nut and is controlled with the help of a calibrated scale.

Moment of friction is changed by a pendulum force-meter consisting of a body, a pair of cogwheels, and a finger with a load or without it.

Depending on the value of friction moment the pendulum is deflected from a vertical position at a definite angle. The value of friction moment is determined by one of 4 scales plotted on a detachable rule.

The tested oil in a quantity of 200 ml, which is necessary for carrying out the test, is poured into an oil glass, where with the help of an electrical stove it is heated to 90° and fed into the oil compartment in which the lower race is rotating. From the compartment the oil with the help of gear pump, having a drive from the shaft of the friction machine, again enters the glass. On the oil line there are cocks with the help of which a constant level of oil is maintained in the compartment. The temperature of the oil which arrives on the friction pairs always remains constant and is measured with the help of a thermocouple connected to a potentiometer.

The outer diameter of the radial-thrust bearings was 47 mm. In the case of the fixed upper sample and a rate of rotation of 200 r/min for the lower sample the rate of slip is:

$$v = \frac{\pi d n}{60} = 0{,}00246\ n = 0{,}00246 \cdot 200 = 0{,}492\ \text{m/s}.$$

The basic index of antiwear properties of oil is the volume or weight wear in a unit of time.

During friction of the mobile lower ring relative to the fixed upper on the surface of the upper ring a hole is formed, the dimensions of which, other things being equal, depend on the quality of lubricating oil used, the loads acting on the ring during their friction, rate of slip, depending in turn on the number of revolutions of the lower ring, and on the quality of the surface of the friction rings. Appraisal of volume or weight wear based on width of friction track will not reflect the real picture of wear, since the quantity of metal removed during friction is a nonlinear function of width of friction track. Actually, the weight of metal removed from the friction surface is determined by the volume included between the two surfaces of the cylinders, and will be equal to the height of

the cylindrical element multiplied by the doubled area of the circular segment, subtended by chord S.

For the circular segment we have:

$$F' = \frac{1}{2}[r(l-s) + sh],*$$

where F' — area of circular segment; r — radius of circumference; ϕ — central angle; h — pointer $h = 2r(\sin\varphi/2)^2$; s — length of chord, subtending arc l with central angle ϕ; $s = 2\sqrt{h(2r-h)}$; l — length of arc, $l = 0{,}0174532\varphi^{\circ} = \sqrt{s^2 + \frac{16}{3}h^2}$.

Inasmuch as the volume of metal v removed from the surface of the ring, which is of interest to us, is determined by the product of the doubled area of circular segment by the length of the cylindrical element, we will have:

$$v = F\cdot a = 2F'a = [r(l-s) + sh]\,a,$$

from which the weight G of removed metal will be equal to $G = v\cdot\gamma$ (γ — volume weight of material of the rings).

Taking γ = 7.85 kg/cm^3 and knowing that 2r = 47 mm, a = 10 mm, it is possible to calculate the weight of metal removed from the surface of ring during friction depending on the width of friction track s.

Thus, having obtained from a series of tests the dependence of weight of metal removed from the friction surface on time with a constant load (P = 150 kg), we will find the basic parameters determining the nature of this dependence.

*[Translators note: one of the "s" quantities in the above equation should probably be capitalized; however, to avoid further confusion, they will be shown as they appeared in the original document.]

With sufficient running in of the tested samples the dependence of weight wear on time is expressed by the equation of a straight line: $y = ax + s_0$. In this case tangent of slope a of the line will constitute intensity of wear, and the value of constant s_0 will characterize certain initial wear.

Treatment of experimental data. If one were to designate: g — wear of rings, mg; t — time of test, h; k — intensity of wear, mg/h; g_0 — initial wear, mg, then it is possible to present change g depending on t in the form

$$g = kt + g_0,$$

where t based on conditions of the test takes the values: $t_1 = 1$ min, $t_2 = 10$, $t_3 = 20$, $t_4 = 30$, $t_5 = 40$, $t_6 = 50$, $t_7 = 60$ min.
Here we will have the values: g_1, g_2, g_3, g_4, g_5, g_6, and g_7.
Using the method of least squares, we obtain:

$$\begin{gathered} \Sigma gt + k\Sigma |t|^2 - g\Sigma t = 0, \\ \Sigma g - k\Sigma t - g_0 s = 0, \end{gathered}$$

where S — number of tests.
From this:

$$\begin{gathered} k = \frac{S\Sigma tg - \Sigma t\Sigma g}{S\Sigma |t|^2 - |\Sigma t|^2}, \\ g_0 = \frac{\Sigma g\,\Sigma |t|^2 - \Sigma t\Sigma tg}{S\Sigma |t|^2 - |\Sigma t|^2}. \end{gathered}$$

Placing in the resulting expressions the known values ($S = 7$, $\Sigma t = 211$, $\Sigma|t|^2 = 9101$) for the given test conditions, after the corresponding conversions we obtain:

$$k = \frac{\Sigma gt - 30\Sigma g}{2741} \qquad (21)$$

$$g_0 = -\frac{\Sigma g}{7} - 30k = 1{,}31\Sigma g - \frac{\Sigma gt}{90}. \qquad (22)$$

Wear of the ring d_0, corresponding to initial wear g_0, obviously will constitute a certain conditional value of friction track at the beginning of application of load P. Therefore the ratio of this load to area $F = d_0 l$ (l – width of ring) will give an index, characterizing the initial carrying capacity of the lubricating film σ_0 with the given load

$$\sigma_0 = \frac{P}{d_0 l}. \qquad (23)$$

Thus the determination of the basic indices of lubricating capacity by the above-stated method is done in the following way.

The oil in question is subjected to 7 tests with a duration of t = 1.10, 20, 30, 40, 50, and 60 minutes. Load on the friction races comprises 150 kg, and number of turns of the lower race – 200 r/min.

Based on width of friction tracks the values of g are calculated after each test and for 7 tests Σg and Σgt are determined. Then by formulas (21) and (22) the values of k and g_0 are determined. The value of initial wear d_0 corresponding to the latter is placed in formula (23), as a result of which σ_0 is found.

3. Method of Tests for Antiburr and Antiwear Properties on a Friction Machine with Area Contact

In accordance with the classification proposed by R. M. Matveyevskiy [110], friction machines with area contact of surfaces include machines where testing samples are fulfilled in the form of elements of a kinematic pair of the sliding bearing type. In a special case this can be a cylindrical shaft and part of a bearing pressed to it from one or from two sides. When machines of such a kind are used nominal specific pressures of an order of 800-1000 kg/cm^2 are realized on the projection of the support.

For tests of lubricating oils with additives on friction machine of such a kind at the IKhP of the Azerbaydzhan SSR Academy of Sciences a method was developed which provides for the appraisal of limits of stability of a lubricating layer against jamming of rubbing surfaces and their relative wear in the process of friction.

Tests were conducted on an LTTO friction machine, a general view of which is shown in Fig. 17. The friction unit and load device are secured on a bed. Through a gear box and a V-belt transmission a driving electric motor imparts rotation to the spindle of the friction unit. Figure 18 shows a kinematic diagram of the machine. As can be seen from this diagram, the spindle of the friction unit through belt transmission $i_1 = 1/4$, removable cylindrical gears, bevel gear $z_7 = z_8 = 30$ and belt transmission $i = 1:2.7$ imparts rotation to a central rod.

Fig. 17. General view of friction machine. 1 – friction unit; 2 – load device; 3 – bed; 4 – gear box.

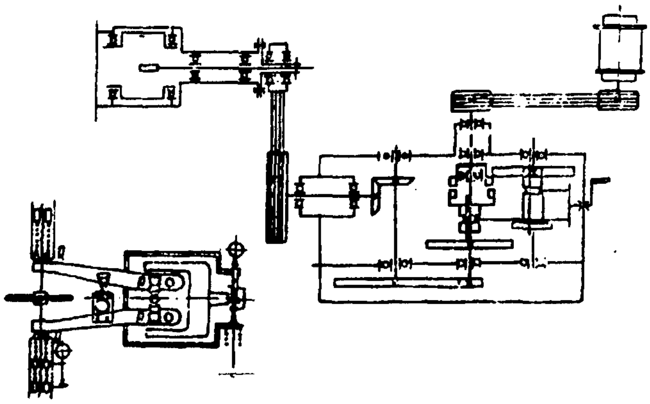


Fig. 18. Kinematic diagram of friction machine.

By means of a step change in the number of revolutions of the spindle of the friction machine (n_1 = 122, n_2 = 258, n_3 = 528, n_4 = 955, n_5 = 2022, n_6 = 4155) it is possible to obtain the following velocities (if the central rod has a nominal diameter d = 1.8 cm): v_1 = 0.115; v_2 = 0.243; v_3 = 0.497; v_4 = 0.900; v_5 = 1.050; v_6 = 3.92 m/s.

Fig. 19. Mechanism for change of transmissions. 1 – removable gears; 2 – lever for switching of transmissions.

Selection of the necessary velocities is carried out by installation of removable gears (Fig. 19) and change of transmissions. The friction unit, a general view of which is shown in Fig. 20, and construction in Fig. 21, constitutes a system, with the help of which friction is made between two cylindrical blocks made from different materials on a central cylindrical rod.

Fig. 20. Friction unit.

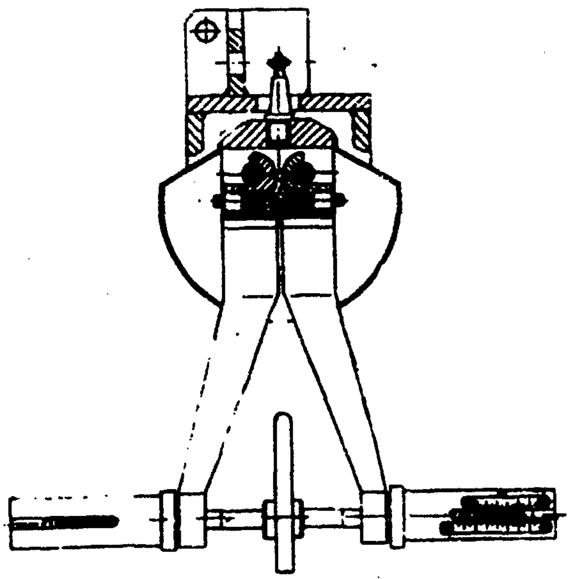


Fig. 21. Construction of friction unit.

Load is applied with the help of a spring load attachment (Fig. 22). Load on the test friction pairs is regulated by selection of springs, the calibration graphs of which are shown in Figs. 23 and 24. For appraisal of wear upon the approach of test blocks in the process of friction the indicators shown in Fig. 25 can be used. By using the calibration graph (Fig. 26) it is possible to calculate the actual load on test samples, and also the specific pressure relative to the area of cross section of a sample with a diameter of 8 mm. Moment of friction is determined by the calibration graph shown in Fig. 27.


a)


b)

Fig. 22. Spring load attachment.
a) levers with loader; b) components of loader.

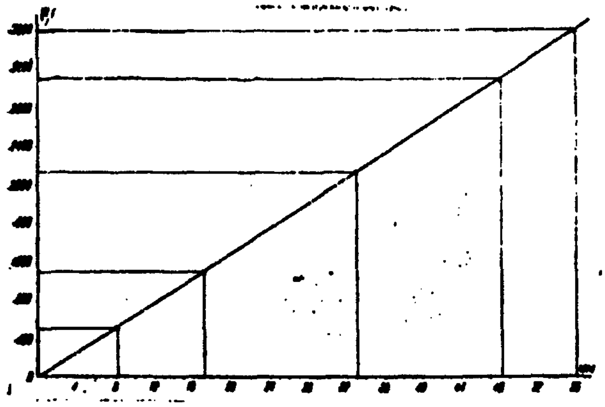


Fig. 23. Calibration curve of a spring with a diameter of 2.2 mm.

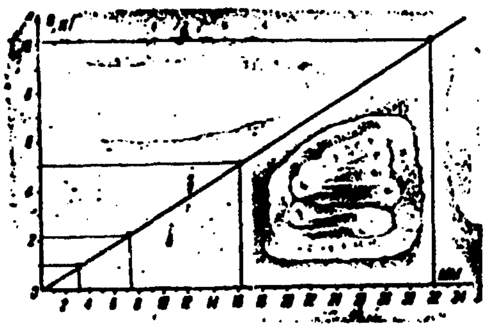


Fig. 24. Calibration curve of a spring with a diameter of 3.5 mm.

Fig. 25. Wear indicators for a friction pair.

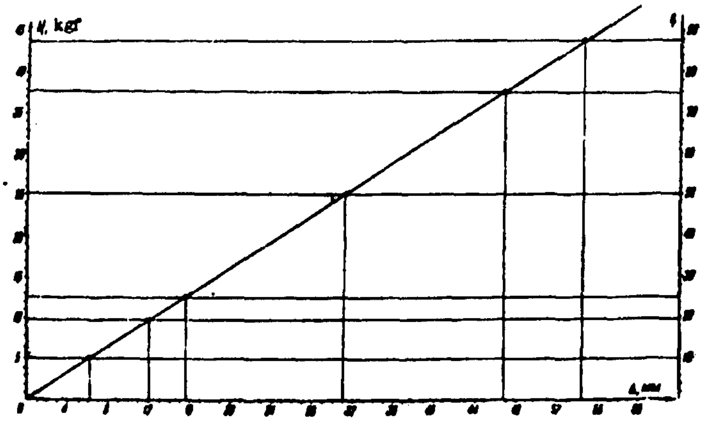


Fig. 26. Calibration graph of dependence of normal load on samples (N, kg) and specific pressure (q, kg/cm^2), relative to the area of cross section of the sample d = 0.8 cm on the scale readings of a spring dynamometer (Δ, mm) (for a spring with a diameter of 2.2 mm).

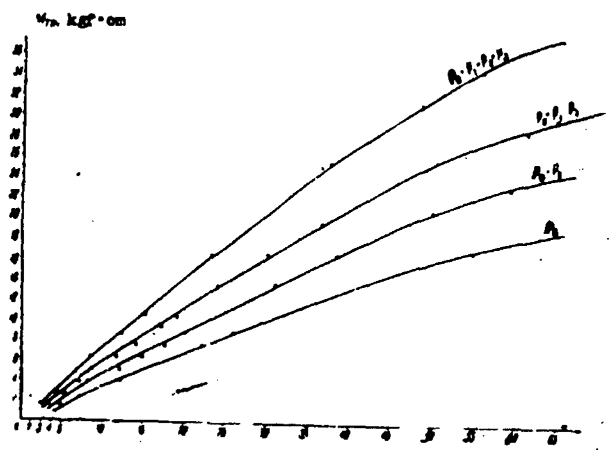


Fig. 27. Calibration graph for moment of friction.

Coefficient of friction is determined by the formula $f = \frac{M_{\text{тр}}}{ND}$, where $M_{\text{тр}}$ – moment of friction kgf·cm; N – normal pressure on samples, kg; D – diameter of central rod (D = 1.8 cm).

For convenience of calculation the combined diagram shown in Fig. 28 is used.

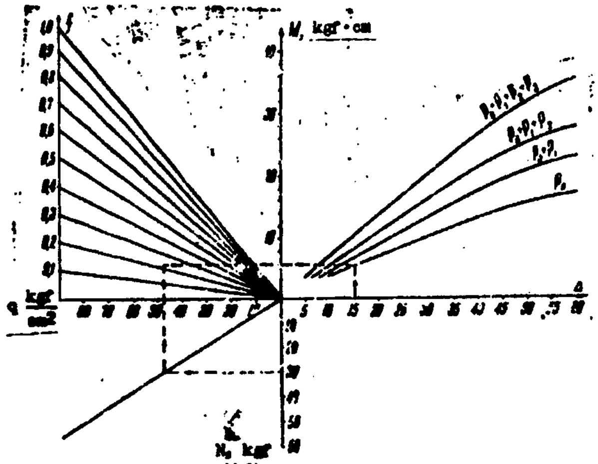


Fig. 28. Combined diagram for calculation of coefficient of friction.

4. Method for Appraisal of the Influence of Quality of Additives on Pitting with Use of Roller Samples

One of the most widespread forms of breakdowns of contacting surfaces of teeth of various type in gear transmissions is pitting or pock-marked breakdown of working sections of teeth profiles.

The opinion exists that the basic cause of these breakdowns is contact fatigue of the material. Operational experience with various kinds of gear trains and also a number of investigations indicate a connection between pitting breakdowns and quality of lubricant used. The direct appraisal of influence of quality of lubricant on pitting which is utilized in practice at a number of research institutes and automobile plants for selection of sorts of oils and additives for gear trains cannot be considered convenient for research works in view of its considerable complexity. In connection with this the mission was assigned to develop a simpler method for appraisal of the influence of quality of oils and additives on pitting.

Without analyzing the numerous hypotheses [30, 96, 116-122] about causes of the appearance of pitting, we will assume that pitting is a phenomenon connected with contact fatigue of the surface material.

Contact fatigue of material, as is known, possesses in general the same characteristic properties and criteria as other forms of fatigue breakdowns. Thus, during tests of materials for contact fatigue curves of fatigue are obtained which have the usual form, which in logarithmic coordinates represent the linear dependence of specific pressures on contact on the number cycles.

Thus the overall number of cycles of stresses N for a case of fatigue breakdown is inversely proportional to the greatest contact stress q in the degree m:

$$N = C \cdot \frac{1}{q_0^m}.$$

Consequently, other things being equal, including an identical number of cycles of stresses, fatigue breakdown of the material of a contacting surface will begin earlier, the greater the value of the largest contact stresses.

As testing machine it was proposed to use the serial MI-8 friction machine, modernized in such a way that on it was possible to obtain sufficiently rapidly and reliably a differentiation appraisal of the influence of oils and their additives on pitting. Operational experience on this machine using the method of appraisal of antiwear properties of oils with additives which was developed by the Institute of Petrochemical processes of the Azerbaydzhan SSR Academy of Sciences in 1961 showed that reliable results can be obtained when using as the friction pairs the outer races of conical roller bearings.

Prolonged operational experience of this friction machine with the use of the outer races of bearing 7204 made it possible to consider that the advantages obtained here can be realized with tests, the goal of which is an appraisal of the influence of quality of oils and additives on pitting. These advantages are reduced basically to the following:

during installation, due to the high accuracy with which the conical inner and cylindrical outer surfaces of the race are machined, beat and misalignments are practically absent;

hardness of outer contacting surfaces changes in very narrow limits;

the outer surfaces are ground under strictly identical conditions, as a result of which practically identical microgeometry of the surface is attained.

Thus as one of the contacting components we selected a cylindrical race. As the second contacting component we selected rings with toroid and cylindrical surfaces. Figure 29 shows a diagram of contact of a ring, having a toroid surface, with a cylindrical ring.

In connection with the presence of residual deformations of the upper roller and the appearance of a friction track on it, calculation of maximum value of contact stresses was performed in the same manner as in a case of contact of two cylindrical surfaces according to the theory of Hertz [38].

For carrying out of tests for an appraisal of the influence of quality of oils on antipitting resistance of materials of cogwheels samples were designed and made which simulated contact of the following pairs of materials: steel on steel, steel on cast iron, bronze on steel.

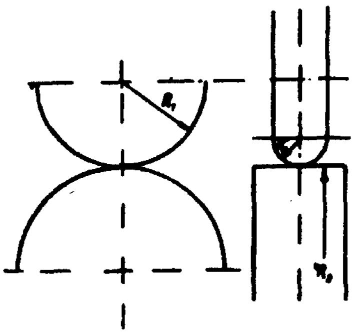


Fig. 29. Diagram contact of toroid and cylindrical rings.

For the first case the most expedient was construction of the sample shown in Fig. 30a. Here the contact of the cylindrical surface of the lower roller with the toroid surface of the upper roller is the most exact and a rapid running in of roller surfaces is attained. Bronze and cast-iron rings, the construction of which is shown in Fig. 30b, are intended for upper rollers, and cast-iron rings (Fig. 30c) – for lower.

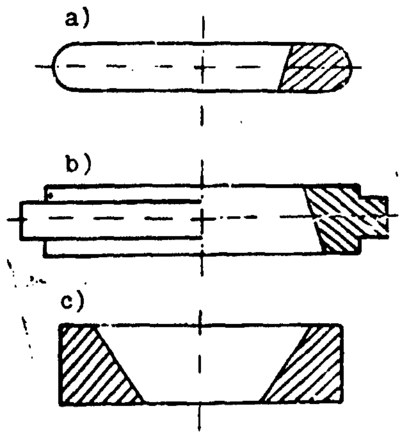


Fig. 30. Roller samples. a) steel; b) bronze and cast-iron; c) cast-iron.

On the surface of the rings beads are made which should perceive the stresses which appear during contact. The value of these stresses, determined according to the theory of Hertz [38] for the possibility of differentation of oils based on their antipitting properties, should be higher than the limit of fatigue crumbling P_{κ}.

The values of limit of fatigue crumbling are usually connected with the hardness of the material from which the test samples are prepared.

Nieman [123, 124] investigated the contact strength of various materials under conditions of frictional rolling of rollers, one of which was leading and worked in a pair with the hardened material of a roller bearing race. Taking as the comparative criterion the coefficient $C = \frac{P_{\kappa}}{H_{\kappa}}$ or the ratio of value of limit of

fatigue crumbling to corresponding hardness, Nieman obtained the values of this coefficient for a number of materials:

carbon steel	0.297
alloy steel	0.358
cast steel	0.33-0.34
ordinary cast iron	0.18
alloy cast iron	0.21
bronze of different brands	0.20-0.34

According to M. M. Khrushchov and others [125], for materials of parts used in automotive construction and tractor construction can be accepted the following value of Brinell hardness (HB): for bronzes – 60-80; for cast iron (gray) – 130-210: for steels – 220-400.

Thus, for the cases examined by us it is possible to obtain:

bronze-steel P_{κ} = 12.0-27.2 kg/cm^2,
cast iron-steel P_{κ} = 23.4-37.9 kg/cm^2,
steel-steel P_{κ} = 67.1-122 kg/cm^2.

Proceeding from the data obtained and taking as the calculation diagram the case of contact of a cylinder with a cylinder, one can determine the values of required loads.

According to M. M. Khrushchov and B. V. Gol'd [125], the following values were accepted for constants of elasticity entering into calculation formulas:

Material	Elastic modulus, kg/cm^2 10-6	Poisson's ratio
High-carbon steel	2.0-2.2	0.29
Cast iron	0.86-1.1	0.25
Cast bronze	0.97	0.33
Rolled bronze	1.3	0.33

For selection of duration of tests experimental data were used which we obtained by G. K. Trubin [126] and in accordance with which for preliminary rough calculations with the use of the minimum possible conditional index of progressing crumbling it is recommended:

for soft and medium-hard steel (approximately less than 400 HB), and also for cast iron the conditional bending point is 1-6 million cycles, conditional base 10, better 15 million cycles;

for bronzes the bending point is 3-12 million cycles with a base of 25-30 million cycles;

for highly tempered steels the bending point is 10-15 million cycles with a base of 100 million cycles.

For appraisal of the influence of quality of oils and oils with additives on fatigue crumbling tests were made of bronze-steel, steel-steel, and steel-cast iron pairs. The tests were conducted on a modernized MI-8 machine. A general view of the installation and its kinematic layout are shown in Figs. 31 and 32. As the lubricant we took AS-6 oil with various concentrations of SB-3 additive. The tests were conducted before the beginning of appearance of pitting breakdowns on the contact surface.

Fig. 31. Modernized MI-8 friction machine.

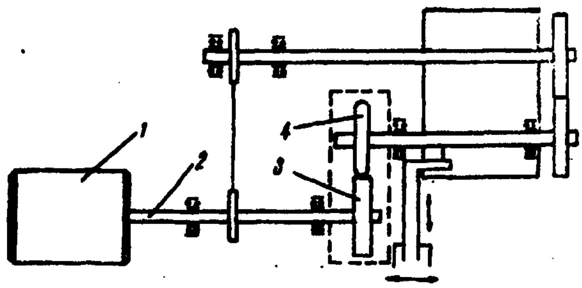


Fig. 32. Kinematic layout of the modernized MI-8 friction machine. 1 – electric motor; 2 – drive shaft; 3, 4 – test samples.

With an increase in the concentration of SB-3 additive in the oil its antipitting stability is considerably reduced. The nature of breakdowns for different metals turns out to be unequal. On the surface of cast iron weakly expressed foci of crumbling appear and these lead to cracks and pock marks.

Bronze rings give intensively developed blisters, which sometimes lead to the formation of breakdowns in the form of a whole track. Pitting breakdowns are clearly observed and the surface of steel rollers also.

5. Method of Appraisal of Carrying Capacity of Lubricating Oils and Oils with Additives on a Testing Machine with Cylindrical Gears

The method for determination of carrying capacity of a lubricating layer or its strength consists of establishing the extreme load for the onset of jamming of working profiles of a pair of cylindrical gears, included in the closed circuit of a test stand of stand under the condition that other indices (temperature of oil, its rate of circulation, rate of rotation of cogwheels, and others) remain constant.

The test stand consists of a machine with a closed circuit with two pairs of cylindrical cogwheels as designed by the Leningrad Polytechnical Institute imeni M. I. Kalinin (LTZK) and two autonomous systems for supply of the tested lubricating material to the testing reduction gears of the closed circuit. The systems for supply of lubricating material are equipped with a device for adjustment and control of the temperature of the tested lubricating oil.

Brief technical characteristics of a machine with a closed circuit: maximum torque in the closed circuit – 180 kg-m; maximum number of revolutions of gears – 1500 a minute (adjustment – stepless); interaxial distance between gears – constant and equal to 144 ± 0.05; with a gear ratio i = 1 the moduli of the cogwheels comprise 2-8 mm; maximum width of tested gears – 60 mm.

A general view of the machine is shown in Fig. 33.

Fig. 33. General view of the machine.

The stator of a beam engine rests on two ball spherical bearings, mounted in steel stands. The latter are rigidly connected with a steel welded frame. Boxes B and V are monolithic constructions. In them the possibility is provided for the rapid installation and dismantling of gears. The shaft bearings for the gears are doubled radial ball bearings. The shaft bearings are radial-thrust ball

bearings, which are intended mainly for reception of axial loads during shifting of the drum along the shafts.

Before the beginning of the test two pairs of new gears are mounted on the machine. They are preliminarily washed in B-70 gasoline or white spirit. After that the autonomous lubrication systems are washed twice — at first with white spirit, and then with the tested oil. After washing the oil is run off. When carrying out a repeated experiment on the same oil it is changed without preliminary washing of the system.

The tests are conducted in the following order. The circulation of oil is started, and the electric motor turned on. The number of revolution is brought up to 1000 ± 10 r/min. Initial load on the lever is set for 10 kg. After a temperature within the limits of 75-80° is established in both reduction gears of the closed circuit the load is increased by 5 kilograms. After a 5-minute run the value of the load on the lever of the balance device is fixed with an accuracy of 0.1 kilograms. After that the load is repeated until the approach of a burr on the working surface of the tooth, which is determined by a sharp increase in friction moment.

Upon completion of the test the load device is readjusted for reverse load of the opposite profile, after which the test conditions are repeated.

For establishing the indices of appraisal of oil quality a load is fixed which corresponds to the start of a burr on the working surfaces of the teeth during direct and reverse loads, and the arithmetic mean value is determined for the load which corresponds to maximum strength (or carrying capacity) of the lubricating layer. The comparative appraisal of carrying capacity when using lubricating oils with various additives shows that with the introduction of the latter the value of the load, at which a sharp jump of friction moment is observed, and on the surface of working sections of profiles of toothed pairs traces of burrs appear, is increased considerably.

6. Method of Appraisal of Antipitting Properties of Oils on an LTZK Stand with a Closed Contour

The practice of exploitation of widely used spur pinions shows that one of the most frequent forms of breakdowns encountered is pitting, the cause of which is accepted as contact fatigue of the material.

It is also known that lubricating oils, being different in the level of viscosity, chemical composition, and also alloyed with various additives, can to a greater or lesser degree affect the efficiency of the gears, determined by their stability to pitting. The goal of the tests described is the appraisal of the influence of oil quality on their antipitting properties. For carrying out the tests an LTZK stand with closed circuit was used. A description of it was given above.

The LTZK stand, on which two independent circulatory oil contours are created, ensures the supporting of assigned temperature rate in both reduction gears. In the preparation of the stand for testing the antipitting properties of oils a bilateral load on tooth profiles is used which permits a comparative appraisal to be made of two samples of oils on one pair of gears without dismantling them. Such an order of tests permits excluding the influence of inaccuracies, which can take place during assembly, on the results of the tests.

Determination of conditions of tests for fatigue breakdown. Contact fatigue of material is characterized mainly by the same peculiarities and dependences which take place for other forms of fatigue breakdowns. It is known, for example, that during tests of materials for contact fatigue the curves of fatigue are similar to the usual type of these curves.

The criteria for appraisal of oil quality from the point of view of their influence on intensity of fatigue breakdowns can be the maximum number of cycles of breakdowns and the limiting value of contact stresses. In first case with constant value of the greatest

contact stresses the tests can be conducted with variable values of number of cycles, and in the second – for every sample of oil it is necessary to take the data for production of the entire curve of fatigue breakdown. Inasmuch as second method is more labor-consuming and prolonged, for the comparative tests we accepted the method of appraisal of antipitting resistance based on maximum number of cycles.

For the assigned number of revolutions at which the test of oil is conducted the overall number of cycles is determined by the total number of revolutions, or $\Sigma N = 60 \cdot T \cdot n$, where ΣN – total number of cycles, T – duration of test (h), n – number of revolutions of gears per minute.

The maximum value of magnitude of contact stresses was calculated for case when LTZK gears with a minimum width of rim of 2 cm are used. Their value at $M_{кр}$ = 60 kg-m turned out to be equal to 13,000 kg/cm^2.

Thus as the basic load parameter the value $M_{кр\ max}$ = 60 kg-m is accepted, and for the appraisal parameter during a comparison of antipitting properties of oils and oils with additives – the value of maximum number of cycles of loads ΣN before the beginning of crumbling on profiles of interlinked cogwheels.

Carrying out of tests and comparison of results obtained with operational data. Since the construction of the stand made it possible to conduct the tests of oils with bilateral load on the profile of the tooth, they included the following stages: running in of gears after installation, test on one of the profiles, test on the opposite profile.

Two samples of oils were tested – AK-15 and nigrol based on GOST 542-50. Selection of these oils was determined by the fact that they are essentially different in their qualities both in initial raw material and also in terms of purification, and therefore one could expect sufficiently apparent difference in their antipitting resistance.

Figure 34 shows a photo of LTZK gears after their tests on these oils. The onset of pitting on nigrol corresponded to $3.6 \cdot 10^6$ cycles, and on AK-15 oil – $6.0 \cdot 10^6$ cycles.

a) b)

Fig. 34. Pitting on teeth of LTZK gears after work, a) on nigrol (GOST 542-50) and b) on AK-15 oil (TU8-61).

The data obtained were compared with the results of tests of transmission oils AK-15 and nigrol (summer) based on GOST 542-50, which were conducted under operational conditions (Table 10).

Table 10. Results of performance tests of nigrol and AK-15.

Appraisal indices	Nigrol	AK-15
Number of breakdowns of surfaces of transmission gear teeth		
in the initial stage with an area of 3 mm^2	78	22
crumbling with an area greater than 3 mm^2	42	92
damage of surfaces at a depth greater than than 0.3 mm	5	5
Number of pittings and crumblings on surfaces of transmission gear teeth	19	none

As can be seen from data in the table, the quality of nigrol from the point of view of influence on fatigue crumblings of gears is lower than AK-15 oil.

These findings obtained as a result of performance tests, are found in satisfactory correspondence with the appraisal of antipitting properties of the same samples when performed on the method developed for bench tests on an LTZK stand.

CHAPTER V

DEVELOPMENT OF COMPOSITIONS TO OILS USED FOR THE LUBRICATION OF INDUSTRIAL EQUIPMENT

Resolution of the problem of ensuring the required level of alloying of oil, used for the lubrication of some form of industrial equipment, is reduced, as was shown above, to the following basic stages. A structural analysis is made of an operational or planned mechanism as well as a kinematic and power calculation of it. In the presence of statistical data on results of exploitation or with an analytic determination of basic parameters of durability separate subassemblies the necessary level of alloying of the lubricant for a given machine is established.

Concrete requirements for the quality of additives used in each case are ensuring definite numerical ratios, emanating from the characteristics of any maximum states of separate subassemblies and machine components.

Having a complex of additives possessing known functional properties, or compositions of them, on model installations the possibility is determined for ensuring the level of alloying which is required for the particular conditions of application of the lubricant. Finally, the concluding stage is carrying out of stand and performance tests on the actual machines.

1. The Use of Additives For Lubrication of Reduction Gears of Molder-Vulcanizers

Investigation of conditions of work of a lubricant in the reduction gear 450-55" molder and requirements presented for the quality of the oil. As is known, work gear drives, representing a variety toothed-spiral gear drives, are characterized by the fact that their initial surfaces not only run in mutually, but also slip relative to each other.

Longitudinal slip of profiles of worm pairs creates unfavorable conditions for their lubrication, since additional slip along contact lines, put together with profile, increases absolute value of rate of slip, which in combination with high loads leads to boundary conditions of lubrication. Here there is the danger of scores, which disturbs the normal work of the unit on the whole.

During operation of worm pairs on reduction gears of the 450-55" molder-vulcanizers (Fig. 35) in separate cases the conditions of normal work were disturbed by increased wear of profile of worm wheel up to sharpening of the tooth; formation of a network of cracks on the surface of the worm, a sharp increase in the power of friction (due to increase of current intensity in the circuit of the driving electric motor) and dragging of the bronze of the wheel on the steel of the worm. With such a kind of maximum states the basic requirement for quality of the lubricant is a high lubricating ability in combination with satisfactory antiburr properties.

The most effective method of ensuring reliable lubrication is the introduction of additives, which, on the one hand, should possess a high degree of surface activity, thanks to which a durable oil layer is created on the friction surfaces, and on the other – should contain chemically active components, which, by entering into an interaction with the metal of the friction surfaces at high temperatures, will form eutectic layers which prevent burrs. Active components of such kind additives, as practice has shown, are sulfur, chlorine, and phosphorus which are found in various compounds.

The conditions of work of a lubricant in a reduction gear of mold-vulcanizers have an inherent number of peculiarities, chief of which is the entry of water into the crankcase of the reduction gear. This water is the product of condensation of vapor used in the system of vulcanization and penetrates there through bearings of the worm gear. The entry of moisture into the lubricant leads to the formation of acid compounds as a result of the interaction of water with chlorine and sulfur compounds which are found in the composition of the additives.

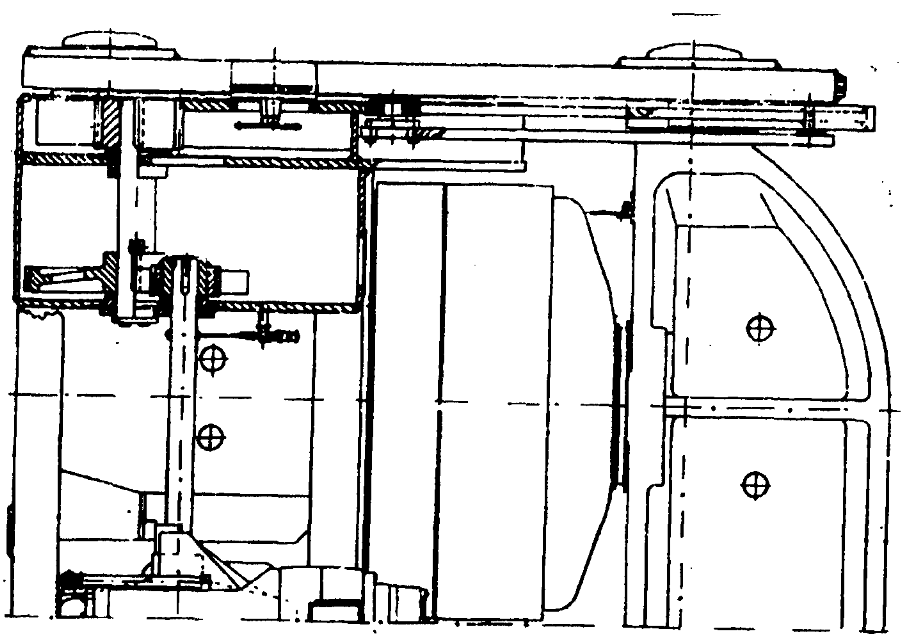


Fig. 35. Molder-Vulcanizer BAG-O-MATIK 450-55".

Furthermore, the addition of chemically active antiburr additives to oil usually creates the danger of intense corrosional wear of the metallic surfaces. Such a kind wear is particularly dangerous for a steel-bronze pair (the latter is the most sensitive to corrosion).

Preliminary selection tests of oils for reduction gears of the 450-55" molder. For establishing the degree of conformity of separate functional properties of a lubricant for reduction gears of molder-vulcanizers to their working conditions in the reduction gear the base oils (bright stock and cylinder-6) in a mixture with the additive recommended for them were tested on laboratory installations, in which were the actual conditions of work of the lubricant in the reduction gear modelled. Lubricating capacity was evaluated on a four-ball friction machine, antiwear properties on the MI-8 friction machine, potential corrosion based on the NAMI method on a DK-2 apparatus, and thermal stability after 50-hour oxidation during 200° in a DK-2 apparatus.

The conditions of friction, at which the lubricating capacity of oils was evaluated on the four-ball friction machine, embrace a wide range of loads, however, based on velocity criteria the conditions of work of the lubricating layer here differ from actual to a considerable degree. As calculations showed the rate of slip in a worm gear, determined by the formula

$$v_{ck} = \frac{mn_r}{19100}\sqrt{z_r^2 + q_n^2} \text{ cm/s [127],}$$

has the values:

a) For the molder-vulcanizer 55"; n_r = 960 r/min (number of revolutions of the worm); z_r = 1 (number of passes of the worm); m – modulus in (mm);

$$m = \frac{t_s}{\pi} = \frac{19{,}95}{3{,}14} = 6{,}3 \text{ мм}$$

(t_s = 19.95 mm – twist of helix of the worm); q – number of moduli in diameter of pitch circle,

$$q = \frac{d_{gr}}{m} = \frac{88{,}9}{6{,}3} = 14{,}1$$

(d_{gr} = 88,9 мм – diameter of the dividing cylinder of the worm);

then $v_{ck} = \frac{6,3 \cdot 960}{19100} \sqrt{1 + 199,81} = 4,4$ m/s.

b) For the moduler-vulcanizer 40": $Z_r = 1$, $t_B = 15.96$ mm and

$$m = \frac{t_n}{\pi} = \frac{15,96}{3,14} = 5,082 \text{ мм};$$

$d_{gr} = 66.9$ mm and

$$q = \frac{d_{gr}}{m} = \frac{66,9}{5,082} = 13,1;$$

$$\text{then } v_{ck} = \frac{5,082 \cdot 960}{19\ 100} \sqrt{1 + 171,61} = 3,35 \text{ m/s}.$$

Proceeding from the calculated speed parameters the corresponding conditions of tests were planned for the MI-8 machine. Determination of potential corrosion according to the NAMI method on a DK-2 apparatus and thermal stability under conditions of oxidation by air was conducted for the qualitative characteristics of relative level of influence of lubricating compositions with additives on corrosion effect and appraisal of their stability during oxidation.

Results of the tests are given in Tables 11-14.

As can be seen from data in the tables, all the test additives improve the lubricating and antiwear properties of oils. The best results are given by sulphurized variants of KhNFK additive added both to cylinder oil and also to bright stock.

Potential corrosion of mixtures of cylinder oil with additives turns out to be higher than for the same mixtures with bright stock. Such a regularity is also noted for the thermal stability of oils. Here the maximum value of corrosion does not exceed 112.5 g/m^2, and deposit after oxidation is no more than 15.37%.

Table 11. Results of testing of test oils with additives on a four-ball friction machine.

Oil	Diameter of friction, d_r, mm	Maximum axial load P_k, kg	Pressure at the beginning of friction P_n, kg/cm2	Pressure at end of friction with a load		
				P_k	200 kg	300 kg
Bright stock						
without an additive	0,52	94	30850	17700	—	—
with 7% ZhNFK (5% chlorinated naptha + 2% BFK)	0,59	121	33500	18100	13500	12000
with 7% AzNII-9 (7% chlorinated naptha + 3% AzNII-7)	0,58	121	33500	18800	13500	16650
7% ZhNFK (sulphurized) (5% chlorinated naptha + 2% sulphurized BFK)	0,57	120	33500	19500	14100	17350
Cylinder						
without an additive	0,54	40	27900	12500	—	—
with 7% ZhNFK (5% chlorinated naptha + 2% BFK)	0,55	95	30950	16400	12350	14450
with 7% AzNII-9 (70% chlorinated naptha 30% AzNII-7)	0,53	108	32200	20050	11350	13100
7% ZhNFK (sulphurized) (5% chlorinated naptha + 2% sulphurized BFK)	0,59	121	33500	18150	14100	17650

Table 12. Results of testing test oils with additives on a MI-8 friction machine.

Oil	Intensity of wear, mg/h
Bright stock	
without an additive	0,99
with 7% ZhNFK (5% chlorinated naptha + + 2% BFK)	0,575
with 7% AzNII-9 (70% chlorinated naptha + + 30% AzNII-7)	0,539
7% ZhNFK (sulphurized) (5% chlorinated naptha + 2% sulphurized BFK)	0,357
Cylinder	
without an additive	0,75
with 7% ZhNFK (5% chlorinated naptha + + 2% BFK)	0,745
with 7% AzIII-9 (70% chlorinated naptha + + 30% AzNII-7)	1,04
with 7% ZhNFK (sulphurized) (5% chlorinated naptha + 2% sulphurized BFK)	0,35

Table 13. Results of testing test oils on a DK-2 installation (potential corrosion, 10 h).

Oil	Corrosion of lead plates, g/m²
Bright stock	
without an additive	+0,55
with 7% KhNFK (5% chlorinated naptha + + 2% BFK)	+0,15
with 7% AzNII-9 (70% chlorinated naptha + +30% BFK)	+0,7
with 7% KhNFK (sulphurized) (5% chlorinated naptha + 2% sulphurized BFK)	3,95
Cylinder	
without an additive	80,45
with 7% KhNFK (5% chlorinated naptha + + 2% BFK)	112,5
7% Az-NII-9 (70% chlorinated naptha + + 30% AzNII-7)	103,0
7% KhNFK (sulphurized)) (5% chlorinated naptha + 20% sulphurized BFK	110,9

Table 14. Thermal stability of test oils with additives (oxidation in a DK-2 device, 50 h).

Oil	Overall deposit, %
Bright stock	
without an additive	3,06
with 7% KhNFK (5% chlorinated naptha + + 2% BFK)	2,81
with 7% AzNII-9 (70% chlorinated naptha + + 30% BFK)	5,47
with 7% KhNFK (sulphurized) (5% chlorinated naptha + 2% BFK)	5,09
Cylinder	
without an additive	13,30
with 7% KhNFK (5% chlorinated naptha + + 2% BFK)	11,44
7% AzNII-9 (70% chlorinated naptha +30% AzNII-7)	15,37
7% KhNFK (sulphurized) (5% chlorinated naptha + 2% sulphurized BFK)	11,51

Considering the high antiwear properties of a mixture of cylinder oil with the sulphurized variant of KhNFK additive, there was considerable interest in testing it under conditions of work of a reduction gear of a molder-vulcanizer in comparison with bright stock and the usual variant of KhNFK additive. Based on the results of preliminary tests a number of oil samples were selected for carrying out comparative tests on the reduction gears of molder-vulcanizers.

Tests under the conditions of work of reduction gears in molders were conducted on operational reduction gears of molder-vulcanizers on the conveyer of a vulcanization workshop. Reduction gears were selected from presses which were not in operation prior to this. The firm's oil was accepted as standard. Tests were made of bright stock with 7% KhNFK and cylinder oil with sulphurized KhNFK. Operating conditions of the reduction gear depended on the intensity of the industrial process. Characteristics of loading of the test reduction gears in the period of the tests were practically identical.

Appraisal parameters were the efficiency of the worm pair according to the results of the systematic observation of it in the process of tests and the change of the initial characteristics of the physicochemical properties of the lubricant, samples of which were taken each month from the crankcases of reduction gears.

In Tables 15-19 data are given on the change in the physicochemical properties of the test oils in reduction gears.

Table 15. Change in the physicochemical properties of scavenge oil (bright stock) with 7% KhNFK.

Indices	Initial oil	Scavenge
Specific weight	0,9207	0,9189
Viscosity, kinematic, cSt		
at 100°	23,19	25,22
at 50°	190,78	—
Ash content, %	0,26	0,45
Water, %	Отс.	—
Test on ChShM		
d_u	0,50	0,53
P_k	121	96
P_v	33500	30890
σ_k	18200	17800
$\sigma_{[illegible]}$	14900	12000
$\sigma_{[illegible]}$	17400	16000
Corrosion of lead plates based on NAPI method, g/m² (GOST 8245-56)	+0,915	52,6
Acid number	Weakly alkaline	Alkaline

Table 16. Change in the physicochemical properties of scavenge oil (standard).

Indices	Initial oil	Scavenge
Specific weight	0,9549	0,9565
Viscosity, kinematic, cSt		
at 100°	39,58	47,73
at 50°	426,24	446,23
Ash content, %	2,35	2,18
Water, %	0,04	—
Test on ChShM		
$d_{и}$	0,64	0,55
$P_{к}$	107	83
$P_{т}$	32100	29600
$\sigma_{к}$	13650	14400
σ_{300}	4060	5100
σ_{900}	5900	6600
Corrosion of lead plates based on the NAMI method, g/m² (GOST 8245-56)	—	133,5
Sulfur, %	—	1,96
Acid number	—	Alkaline

Table 17. Change in the physicochemical properties of scavenge oils from the crankcase of a reduction gear (cylinder with sulphurized KhNFK).

Indices	Initial oil	Scavenge
Specific weight	[illegible]	[illegible]
Viscosity, kinematic, cSt		
at 100°	34,14	[illegible]
at 50°	355,72	[illegible]
Ash content, %	0,51	[illegible]
Water, %	Отс.	[illegible]
Test on ChShM		
$d_{и}$	0,58	[illegible]
$P_{к}$	[illegible]	[illegible]
$P_{т}$	[illegible]	[illegible]
$\sigma_{к}$	[illegible]	[illegible]
σ_{300}	[illegible]	[illegible]
σ_{900}	[illegible]	[illegible]
Corrosion of lead plates based on the NAMI method, g/m² (GOST 8245-56)	110,9	+5,8
Acid number	Neutral	Weakly alkaline

Table 18. Change in the physicochemical properties of scavenge oil (cylinder with sulphurized KhNFK).

Indices	Initial oil	Scavenge
Specific weight	0,9283	0,9387
Viscosity, kinematic, cSt		
at 100°	34,14	42,98
at 50°	396,37	565,[illegible]5
Ash content, %	0,21	0,34
Water, %	Absent	0,15
Test on ChShM		
$d_{и}$	0,58	0,56
$P_{к}$	121	83
$P_{т}$	33500	29500
$\sigma_{к}$	18800	13800
σ_{300}	14100	—
σ_{500}	17000	—
Corrosion of lead plates based on the NAMI method, g/m^2 (GOST 8245-56)	110,9	7,1

Table 19. Change in the physicochemical properties of scavenge oil (bright stock with KhNFK).

Indices	Initial	Scavenge
Specific weight	0,920	0,9158
Viscosity at 100°		
kinematic, cSt	25,59	27,27
conditional, degree	3,63	3,86
Viscosity at 50°		
kinematic, cSt	219,05	244,24
conditional, degree	29,57	32,97
Ash content, %	0,21	0,25
Water, %	0,06	0,06
Test on ChShM		
$d_{и}$	0,55	0,58
$P_{к}$	120	108
$P_{т}$	33400	32200
$\sigma_{к}$	20700	15700
σ_{300}	15100	7120
σ_{500}	17700	7430
Corrosion of lead plates on a DK-2 device, g/m^2	71,25	2,9

As can be seen from data given in the tables, the basic indices of quality of oil do not undergo considerable changes.

Besides the comparative test samples a series of reduction gears from a vulcanization conveyer were under detailed observation for 5 years for accumulation of data on the results of performance of bright stock with 7% KhNFK additive in reduction gears. The

method of tests and control of their course were the same. The test oil and chlorinated naptha were from plant production. Physicochemical indices of the quality of the test lot of oil are given in Table 20.

Table 20. Physicochemical indices of the quality of a test lot of bright stock with 7% KhNFK.

Indices	Bright stock without an additive	Bright stock with KhNFK additive
Specific weight	0,9035	0,9163
Viscosity, kinematic at 100° cSt	26,64	25,35
Coke, %	0,83	1,46
Ashes, %	0,011	0,17
Water based on Dean and Stark, %	—	0,05—0,055
Acid number	Alkaline	Alkaline
Lubricating properties on ChShM		
$d_и$	0,47	0,58—0,61
$P_к$	44	121—121
$P_т$	23900	33600—32500
$\sigma_к$	10400	18800—17000
σ_{700}	—	13200—12900
σ_{900}	—	16300—16000

This oil ensured the current production needs in a lubricant for reduction gears on molder-vulcanizers on conveyer lines of a vulcanization workshop. Almost all the reduction gears, as one can see from data in Table 21, worked on this oil. Prolonged exploitation of this oil showed that in comparison to oil without an additive, and also with other additives, it ensured the satisfactory performance of reduction gears in both the 55" and the 40" molder-vulcanizers.

Table 21. Type of molder and oil used.

Line of conveyer	Type of molder	Oil used
Row I	55"	INKhP
Row II	55"	INKhP
Row III	40"	Bright stock and cylinder
Row IV	55"	Firm's
Row V	55"	Firm's and INKhP

On the basis of positive operational experience technical specifications have been developed for lubricant for reduction gears on molder-vulcanizers (Table 22).

Table 22. Provisional technical specifications for lubricants for reduction gears on molder-vulcanizers.

Indices of quality	Bright stock with 7% KhNFK (5% chlorinated naptha + 2% BFK)	Cylinder-8 with 7% KhNFK (5% chlorinated naptha + 2% sulphurized BFK)
Viscosity, kinematic, cSt		
at 100°	23,19	34,14
at 50°	190,78	395,77
Ash content, % – no less than	0.26	0.21
Content of water, %	Traces	Traces
Acid number, mg KOH	Weakly-alkaline	Neutral
Corrosion of steel and copper plates based on GOST 2917-45	Absent	Absent
Corrosion of steel plates in humid medium	Absent	Absent
Test on ChShM		
d_{u}	0,59	0,59
P_{k}	121	121
P_{r}	33500	33500
e_{k}	18100	18150
e_{50}	13500	14100
e_{100}	12300	17650
Intensity of wear on MI-8 friction machine based on the INKhP method, mg/h – no more than	0,6	0,4

2. The Use of Additives For Lubrication of Reduction Gears with Novikov Engagement

Until recently toothed gears with involute gearing were used exclusively for power drives. With such propagation involute toothed gears have to have inherent well-known merits.

However, involute tooth gears cannot be placed in many drives in the available dimensions (aviation drives, built-in reduction gears of mining machinery, and others). Premature breakdown of cogwheels hampers the exploitation of machines, increases operational expenditures, and in certain cases can lead to accidents.

All of this, and also the constantly increasing power of machines, and from here the strain on their components, require increase of carrying capacity of tooth gears.

It is known that the values of contact stresses decrease with a decrease of the given curvature of two contacting surfaces. For an involute gearing the decrease of given radius of curvature is possible only in the case of internal engagement.

The carrying capacity of gear teeth could be increased considerably if along with external engagement of wheels the teeth had internal contact. For straight-toothed gears with external engagement the realization of convex teeth on one of its wheels and concave on the other led to irregularity of rotation, and therefore such a gear is impracticable. For helical gears such a form of teeth can be realized with observance of the condition of uniformity of rotation.

The problem of developing a toothed gear with high carrying capacity was solved universally by M. L. Novikov [128]. He developed the geometric theory of a fundamentally new engagement for gears with parallel, transverse axes, making it possible to increase by several times the carrying capacity of gears.

The line of action in such a gear is disposed parallel to the axes of the wheels and stands at a certain distance l from the pitch line. Segment l comprises the angle of pressure $d_{д}$ with a common tangent to the pitch circles (Fig. 36). The point of engagement moves along the line of action with a constant speed. In order to ensure uniform rotation, an engagement fact ε = 1.1-1.2 is designated across the length of the wheels. In this case prior to disengagement of one pair of teeth the next pair enters into engagement.

Teeth in the face plane have a circular or close to circular profile. The lateral surface of the teeth is formed by shifting such a face profile in the manner of a helix.

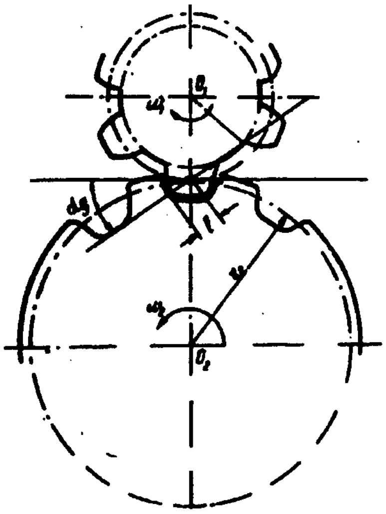


Fig. 36. Diagram of the engagement proposed by M. L. Novikov.

One would think that point contact, taking place at $R_2 > R_1$ (Fig. 37), cannot ensure a high carrying capacity. However, M. L. Novikov theoretically founded a new principle of formation of tooth surfaces and experimentally proved that for a point tooth the elastic zone of contact formed under the impact of load, in the case of the proper selection of parameters, can have a considerable area, as a result of which the carrying capacity of the tooth not only does not decrease, but increases considerably. For obtaining a high carrying capacity for contact stresses the angle of arrival of the teeth is designated as small as possible – within limits of 20-15°.

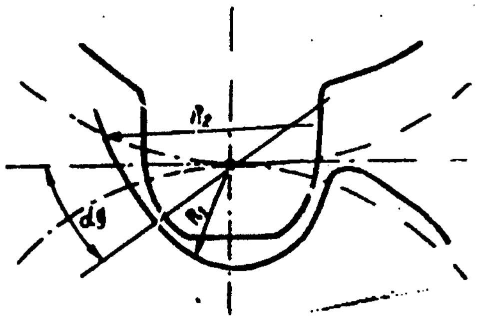


Fig. 37. Contact of tooth profiles in a circular-helical engagement.

The given radii of curvature along contact lines are increased rapidly with a decrease of angle β. The given radii of curvature in the face plane depend on the value of difference of radii of curvature of profiles R_2-R_1. In the process of running in this difference approaches zero, contact becomes linear along the depth of the tooth - the given radius of curvature in the face section is equal to infinity.

Since the radical solution of the problem of increasing the carrying capacity of cogwheels with help of M. L. Novikov's point tooth gearing the idea of developing gears of such a type served as the starting point for various modifications of the M. L. Novikov engagement.

The Novikov gear is a tooth gear with parallel, transverse, and intersecting axes with one line of action, which can be located on various sides from the pitch point. In 1955 R. V. Fedyanin [129] formulated the idea of developing a gear with two lines of action (dozapolyusnyy) [Translator's note: Russian word not identified: do-before; za-after; polyusnyy-pitch; possible breakdown of word].

Extensive experimental checking during the last few years showed that the potential possibilities of the idea of M. L. Novikov are used more fully with dozapolyusnyy variant of engagement. It possesses a greater engagement face, lesser sensitivity to increase of center-to-center distance, greater resistance to fatigue wear, and is simpler in a technological respect [130].

Conditions of Work and Requirements Presented for Quality of a Lubricant for Circular-Helical Engagement (Novikov)

In spur gears with Novikov engagement the working helical surfaces of the teeth (having a circular profile in normal section) during turning of the wheels roll over each other theoretically without sliding. The point of contact travelling along the tooth is

always at the same distance from the geometric axes of the wheels. The velocity of shift of contact along the tooth and pressure angle always remain constant. After sufficient running in contact is realized almost on the entire depth of the tooth, and in length – up to 1/3 of the tooth.

Due to the high rate of longitudinal rolling of the gear-contact pattern in a Novikov engagement theoretically conditions are created for the formation of a constant carrying oil layer between the teeth.

Thus, conditions favoring liquid lubricants predominate here to a greater degree than this takes place in involute gears. Hence the smaller friction coefficients with other things being equal, which is connected with lesser liberation of heat. The latter ensures a lesser tendency to jamming and burrs.

In summing up the findings obtained by various researchers it is possible to point out the following advantages of Novikov gears as compared to involute.

1. With identical parameters gears Novikov with HB < 350 ensure approximately a double increase of load capacity, determined by contact strength [131, 132], a double decrease of losses in engagement [133], and 3-6 times less wear [134].

2. For a Novikov engagement the value of given radius of curvature comprises an order of 100 m, and for involute – 2-3 m.

3. Smaller coefficients of friction, ensuring a smaller increase of temperature and liberation of heat. These factors, and also more favorable conditions for the creation of hydrodynamic conditions of lubrication, promote the elimination of jammings. However, in a Novikov engagement the running in of wheels becomes of paramount importance, and conditions of normal operation depend more on the quality of breaking in here than, for example, with an involute engagement.

Operational experience with dozapolyusnyy gears [130] showed that in the period of running in light burrs and flow of metal are observed. In the process of operation traces of scoring and fatigue breakdown are eliminated. A high degree of resistance of these gears to pitting is observed. If, for example, when working on involute gears made from st. 45 (HB = 170-241) pitting is observed after 2-4 million cycles, then with working on Novikov wheels with dozapolyusnyy engagement stability is increased up to 10-12 million cycles [135].

V. P. Kudryavtsev [136] considers that in Novikov gearing the time that the point of the tooth surface remains in the zone of contact is greater than for involute, therefore the danger of gripping appears. Consequently, a lubricant possessing antiscoring properties is necessary.

A. N. Grubin [137] points out that the intensity of crumbling increases with a lowering of viscosity and with an increase in the quantity of lubricant. Proceeding from this, one should assume that increasing of viscosity is an effective means of ensuring the reliable work of gearing of this type. However, A. N. Grubin makes an opposite conclusion. The usual recommendations on the selection of lubricating oils for tooth gears [138] anticipate the necessity of decreasing the viscosity of oil with an increase of velocity and increasing it with an increase of pressure.

Viscosity also has a significant influence on the value of friction coefficient on the working surfaces of tooth profiles [139], and consequently on the value of efficiency.

Regarding the influence of viscosity on the time before the beginning of breakdown due to contact fatigue, then here there are data on more favorable conditions with the use of high-viscosity oils [123].

Low-viscosity oils, along with an increase of velocity of the wheels, can also ensure intense heat exchange in the reduction gear at the expense of improvement of conditions of circulation [140].

L. I. Tishchenko [141], in investigating the connection between viscosity and wear, established that although IS-50 oil has a viscosity of 1.7 and 5 times lower than that of AK-10 and bright stock, it ensures lesser wear of working tooth surfaces.

V. N. Gersator [142] points out that the factor of viscosity does not determine the load capacity of teeth.

Such contradictory indications with respect to the influence of viscosity on the work of gearing, in our opinion, can be explained by the fact that usually the data obtained include both the results of work of teeth in the region of boundary lubrication and also under conditions when hydrodynamic conditions are still preserved. Consequently the recommendations of Merrit [143] on selection of oils based on viscosity, which were founded on experimental data, are doubtful in general, and all the more so in respect to Novikov wheels.

In the literature there are indications of the expediency of selection of lubricants based on oiliness [144]. For heavily loaded gears in [145] it is recommended to add sulfur, thus increasing the strength of the oil film by 2-3 times.

In work [132], devoted to Novikov engagement, they used heavy lubricants which ensured a considerable thickness of the oil layer on the working surfaces. This is necessary in view of the triple overload of gears as compared to involute engagement.

The limit of safe load for the working of gears with sufficient longevity [146] depends to a considerable degree on the properties of the lubricant to resist the action of thermal outbursts on points of contact with a duration up to 0.001 s and with local increases of temperature up to 500-600°, which break down the lubricant and the friction surface [55].

A serious factor, ensuring the reliable work of a lubricant, should be considered its resistance to oxidation. According to the theory developed by S. V. Ventsel' [147], concerning contact oxidation of lubricating oils during friction, a considerable share of the products of oil contamination during friction are due to the appearance of local outbursts. Usually aging of oils implies not only its oxidation, but also its contamination with impurities which enter the reduction gear from without. These factors essentially lower the period of service of oil in a reduction gear.

Solid particles, getting into the zone of engagement of the wheels, cause an irregularity of rotation [55], as a result of which the requirements of high stability and ensuring of a minimum level of mechanical impurities during the lifetime of the oil in Novikov gearing take on an important meaning.

According to certain authors [148], there are limiting values of viscosity, the exceeding of which hampers the starting of reduction gears. To ensure a sufficient mobility of the lubricant under these conditions extensive use is made of dilution of oil with low-viscosity fractions [149, 150].

Basic Trends in the Alloying of Lubricants for Noviko Gears

For the determination of directions in the development of optimum lubricating compositions, satisfying the conditions of work of Novikov gearing, it is necessary first of all to establish to what degree the addition of additives to a lubrication oil which is used in a reduction gear with Novikov engagement can influence the conditions of work of the gearing.

Proceeding from our analysis of the influence of additives on the change of maximum states, it is possible to examine at least three directions of influence of additives on conditions of contact — antiscoring, determined mainly by its resistance to breakdown with an increase of contact temperatures, antiwear, and antipitting.

Let us consider the basic parameters, characterizing the resistance of a lubricating film to breakdown in the case of an increase of contact temperatures.

According to the method proposed by A. I. Petrusevich and associates [151], as the criterion of breakdown of the oil film between gear teeth with Novikov engagement one can accept the instantaneous increase of temperature ϑ in the zone of contact, determined by the formula of Block

$$\vartheta = \frac{0{,}83}{\sqrt{\lambda\gamma c}} \cdot \frac{q\, v_{ск} f}{\left(\sqrt{v_{ш}} + \sqrt{v_{к}}\right)\sqrt{b_1}}\ ^{\circ}\mathrm{C},$$

where q – specific load along length of contact line, kg/cm; $v_{ск}$ – rate of sliding on tooth profiles, cm/s; f – coefficient of friction; λ – coefficient of thermal conductivity of material of cogwheels, Cal/cms deg; γ – specific weight of wheel material, kg/cm^3; c – specific heat of material, Cal/kg deg; $v_{ш}, v_{к}$ – velocity of rolling (shift of zone of contact along the teeth), cm/s; b_1 – halfwidth of area of contact, cm.

As can be seen from the Block formula, the increase of temperature in the zone of contact in the case of the same values of specific load on the length of the contact line grows with an increase of $v_{ск}$ and f, and decreases with an increase of λ, γ, c, and sum of roots from $v_{ш}$ and $v_{к}$.

Values $v_{ск}$, $v_{ш}$ and $v_{к}$ can be determined from the known kinematic relationships for Novikov gearing. For example, $v_{ск}$ of working wheels is determined:

$$v_{ск} = v\left(\frac{i+1}{i}\right)\bar{e},$$

where $\bar{e}$ – relative displacement, $\bar{e} = \frac{e}{R_H}$, R_H – radius of pitch circle, i – gear ratio.

The velocity of rolling of working surfaces along the contact line:

$$v_{ш} = v\sqrt{1 + tg^2\varphi + \bar{e}^2 + 2\bar{e}\sin\sigma_0},$$

$$v_{к} = v\sqrt{1 + tg^2\varphi + \left(\frac{\bar{e}}{l}\right)^2 - \frac{2\bar{e}}{l}\sin\alpha_g}.$$

The vectors of these velocities form with the line of contact an angle of 70-80°.

In the given formulas, ϕ – angle of ascent of helix of tooth, $\phi = 90-\beta$ (β – angle of inclination of tooth on the dividing cylinder); σ_g – mean value of pressure angle.

The relationship of values of velocities indicated is such that in various points of the tooth their value has a different influence on temperature rise. Furthermore, in various points of the tooth along its whole depth the value of specific load q and also the friction coefficient f change. Maximum increase of temperature in the case of one-sided engagement will be observed on the crest of convex [sic] and on the root of convex teeth.

The safety margin of Novikov gears against disruption of the oil film is also appraised by the safety coefficient for break in oil film:

$$k_0 = \frac{q_0(i+1)}{D_{ш}i} \text{ kg/cm}^2,$$

where q_0 – specific load per unit of width of gear, kg/cm; $D_{ш}$ – diameter of pitch circle, cm.

Coefficient k_0 can be expressed through Φ, considering that for nonrun-in surfaces the friction coefficient is 1.5 times greater than that obtained based on experimental data for run-in surfaces:

$$f = \frac{6}{\rho^{0.375}\cdot\sqrt[4]{\nu v_{ck} v_{к}}},$$

where v_{ck}— rate of sliding on tooth profiles, cm/s; v_t— total velocity of rolling, $v_t = 2\frac{v_{okp}}{tg\,\beta}$ (sum of velocities of shift of zone of contact of tooth surface for Novikov gears, v_t= 20-100 cm/s); v_{okp}— peripheral velocity, cm/s; ν— viscosity of oil, cSt.

Extreme values $\frac{v_{ck}}{v_t}$; which are encountered in gears Novikov comprise 0.0316-0.0373 [152]. The increase of temperature is unequal for different points along the depth of the tooth, since in this direction the specific load q and coefficient of friction f change.

An essential influence is exerted on the conditions of work of the lubricating layer in the zone of contact by the value of the friction coefficient. Therefore, by changing the frictional characteristics of the lubricating layer we can influence the value of critical parameters, determining its stability during the work of engagement.

The influence of additives on the change of frictional properties of lubricating layer can be visually illustrated by the data obtained on the MAST-1 friction machine.

A comparison of antifriction properties of various additives in a wide range of temperatures shows that they can ensure a different effect. From the data in Table 23 one can see for example that already with temperatures of an order of 60° it is possible to observe a difference in antifriction properties of the same quantities of INKhP-30 and INKhP-46 additives. It is natural that the difference in base oils cannot give the same effect as is ensured by the introduction of additives.

In the same way it is possible to speak about the influence of thermal conductivity. Thermal conductivity of the lubricating layer can be determined from the expression:

$$\lambda = 0{,}293\,\frac{\rho^{2,15}\,c^{1,65}\,M^{0,912}}{\mu^{0,12}},$$

where M — molecular weight, μ — viscosity, centipoise.

Table 23. Change in the friction coefficient of 1% solutions of additives in oil.

Oil	Temperature, °C												
	20	40	60	80	100	120	140	160	184	200	220	240	300
Initial without additive	0,09	0,09	0,09	0,09	0,09	0,10	0,10	0,1[illegible]	0,12	0,28	—	—	—
The same oil with additives													
INKhP-30	0,05	0,05	0,06	0,07	0,07	0,09	0,09	0,10	0,10	0,14	—	—	—
INKhP-46	0,05	0,05	0,05	0,05	0,05	0,05	0,05	0,05	0,05	0,05	0,05	0,09	0,06

For ind-50 oil, for example, λ = 0.13, and for nigrol-3 λ = = 0.15 Cal/mh °C. The insignificant difference between values of this constant for base oils is explained on the basis that for oil fractions the change in density is insignificant. The same can be said about the value of molecular weight. Regarding viscosity, then it enters into the expression for determination of λ to the degree of 0.12, which indicates its insignificant influence.

Heat capacity C depending on the operating temperature of the oil and its density is determined in the following way:

$$C = \frac{273+\theta}{288}\left(0{,}7125 - 0{,}31058\,\rho_{60}\right),$$

ρ_{60} — density of oil at 60°,

$$\rho_{60} = \rho_{20} + \gamma\,(60-20),$$

γ — temperature correction, ρ_{20} — density at 20°, θ — operating temperature of oil.

As is known, heat capacity of a substance depends on its atomic heat capacity. A comparison of values of atomic heat capacity for various elements is given in Table 24 [153].

Table 24. Atomic heat capacity for certain solid and liquid substances, cal.

Substances	Elements						
	C	H	N	O	Si	S	P
Solid	1,8	2,3	2,7	4,0	4,8	5,4	5,5
Liquid	2,8	4,3	—	6,0	5,8	7,4	7,0

As can be seen from data in the table, the addition of additives containing sulfur and phosphorus to oil increases the heat capacity of the lubricating layer, which leads to a lowering of its critical temperature.

The value of viscosity entering into the formula of Block obviously should be corrected taking into account the value of pressure which is effective in the lubricating layer on contact between teeth. This can be done in the simplest form, by using the Kiskal't equation. This determines the dependence of change of viscosity on pressure. It has the following form:

$$\rho_p = \rho_0 a^P,$$

where ρ_p – viscosity at pressure P (kg/cm^2), ρ_0 – viscosity at atmospheric pressure. The α – constant, depending on temperature, for example, for petroleum oils α = 1.002-1.004, for glycerine – 1.0005, and for castor oil – 1.001.

The nature of change of viscosity with an increase of oil pressure is represented graphically in Fig. 38. The introduction of considerable quantities of additives will have a significant influence on piezoeffects in boundary lubricating layers.

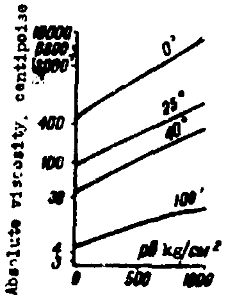


Fig. 38. Dependence of viscosity of oil on pressure at different temperatures.

Dow [154], in investigating the viscosity of mixtures of various liquids with a change of temperature from 30 to 75° and a change of pressure up to 12,000 kg/cm^2, found that certain mixtures are subject to a linear dependence of logarithm of viscosity on composition (rule of Arrhenius), while for a number of mixtures complex deviations from this rule were observed. Anomalous deviations took place both in final and also in intermediate concentrations. This same author [155] treated the data of experiments theoretically, on the basis which it was possible to show that the Bachinskiy theory is inapplicable, since viscosity is not a function only of volume. The results of measurements of viscosity of various oils in a range of pressures up to 4000 kg/cm^2 showed [156] that the chemical composition of oils also exerts a significant influence on piezoeffects. Thus at a pressure of 2000 kg/cm^2 the increase of viscosity for California oil was 4 times greater than for Pennsylvania. Experiments with these oils were subsequently conducted in a range of temperatures of 38–90°C.

In the same range of temperatures Dow, Fenske, and Morgan [157] investigated the viscosity of oils without additives and two chlorinated diphenyls. For one of the diphenyls the greatest of the investigated effects of viscosity was observed – at pressure of all told 250 kg/cm^2 viscosity was increased by 15 times.

Morgan and Dow [158] measured the viscosity of ortho-, meta-, and para- toluenes and their monosubstituted compounds with bromine, iodine, and the nitro group. It was determined that the rate of increase in viscosity with an addition depends on the position of the substituents. Thus, if the atomic weight of the replacing group is increased, then the rate of increase of viscosity with an increase of pressure turns out to be greater for ortho- and meta-positions and decreases for the para-position.

Dibert, Dow, and Fink [159] investigated the viscosity of various Pennsylvania oils in the same range of temperatures and pressures. An increase was observed in the rate of increase of viscosity with pressure with an increase of molecular weight. The temperature coefficient of viscosity with constant pressure also increased with an increase of molecular weight for all values of pressure.

In the tests conducted by Dow, McCartney, and Fink [160] it was possible to establish that with a predominance of napthenic and aromatic hydrocarbons in oils over paraffin pressure exerts a great influence on viscosity of oils. The results of the experiments made it possible to obtain the operational dependence of viscosity on density.

The linear relationship between pressure and logarithm of viscosity was found by Suga [161].

The dependence of complete outflow on pressure can be used for calculation of the derivative of viscosity at any point [74].

Optimum relationships between derivatives of viscosity based on temperature and pressure [162] are practically important for the determination of conditions of work of boundary lubricating layers under conditions of high and superhigh pressures on contact. The connection between lubricating effect and value of the derivative of viscosity based on pressure was investigated by Needs [163]. It was

determined that the effective coefficient of friction has a minimum value at high pressures. The introduction into lubricating compositions of additives which are different in influence on antiscoring properties thus will have a simultaneous effect on a change of dependence of viscosity on pressure. There will be a similar influence on polymeric additives which are used as thickening components. In particular, experimental works are known, in accordance with which it is established that fibrilose molecules, included in the composition of additives of such a type, promote increase in the derivative of viscosity based on pressure [164].

On the basis of this analysis certain conclusions can be made:

a number of physicochemical and mechanical characteristics of additives which are introduced into oils in comparatively small quantities can exert an essential influence on its operating characteristics;

the greatest effect can be achieved by the application of additives of the antifriction type, ensuring lowering of the friction coefficient; as a result of their addition to the lubricant value of instantaneous increase of temperature decreases, This has a positive effect on the maximum efficiency of the boundary layer.

Experimental Investigation of the Influence of a Lubricant on the Operation of Gears with Novikov Engagement

The analysis of conditions of work of a lubricant in this form of engagement, and also the results of observations of the nature of the basic forms of breakdowns of working profiles make it possible to arrange the most frequently encountered causes of these breakdowns into the following sequence: contact fatigue, jamming, wear.

In the first stage of the experimental investigations comparisons were made of the nature of resistance to breakdowns due to contact fatigue of oils intended for use in reduction gears Novikov engagement.

From source material it is known that maximum contact pressures have values of an order of 10,000 kg/cm^2. Therefore in the planning the conditions of the tests this value was accepted as initial.

For studying the influence of lubrication on maximum efficiency of contacting profiles of Novikov engagement test samples were used which were made from material analogous to that used for gears with this engagement.

In the tests the number of cycles was considered before the beginning of breakdowns on contacting surfaces. Table 25 shows the results of the tests conducted.

Table 25. Results of testing of oils.

Samples	Number of cycles before the beginning of breakdown
AK-15, tests	
I	$5{,}4 \cdot 10^5$
II	$5{,}2 \cdot 10^5$
III	$3{,}1 \cdot 10^5$
Average	$4{,}5 \cdot 10^5$
AK-10, tests	
I	$2{,}7 \cdot 10^5$
II	$3{,}5 \cdot 10^5$
Average	$3{,}1 \cdot 10^5$
Cylinder-24 tests	
I	$2{,}85 \cdot 10^5$
II	$2{,}77 \cdot 10^5$
III	$1{,}95 \cdot 10^5$
IV	$3{,}00 \cdot 10^5$
V	$3{,}15 \cdot 10^5$
Average	$2{,}74 \cdot 10^5$

In previous investigations it was established that nigrol and AK-15 give an approximately identical level of pitting on working surfaces of teeth of heavily loaded spur gears. A lowering of the viscosity of oil, as also an increase of it, as can be seen from data in Table 25, gives a certain lowering of longevity.

The determination of antiscoring characteristics of lubricating oils was performed according to the method regulated by GOST 9490-60. The value of maximum load of jamming was determined for various oils. In Table 26 data are given concerning the maximum axial load for various oils. As can be seen from these data, the quality of base oil has little influence on the index of maximum load of jamming.

Table 26. Maximum axial loads.

Oil	Axial load P_0 based on GOST 9490-60 in degrees
Spindle-2	18
Industrial-50	20
Machine S	18
DS-14	20
Cylinder	19
Vapor*	20

Note *Vapor – a steam engine lubricating oil.

The experimental investigation of influence of quality of oil on intensity of wear was conducted on samples of base oils of various viscosity. It was determined that viscosity within the limits of 6-8 cSt at 100° could be considered optimum.

Proceeding from the results obtained, and also considering actual resources of oils for these purposes industrial-45 oil was recommended for further tests.

Appraisal of the effectiveness of alloying this base oil was also conducted from the point of view of the influence of qualities of additives on a change in the limits of contact fatigue of surface material, resistance of lubricating layers to breakdowns, combined with phenomena of jamming, and ensuring the maximum wear resistance of working sections of tooth profiles.

The most effective in this respect turned out to be the INKhP-46 additive. With this additive the time before the beginning of breakdown due to contact fatigue is increased from 21 min (oil without

additive) to 100-120 minutes. The INKhP-46 additive also acts effectively on the change of antifriction properties of the lubricant, ensuring in all ranges of operating temperatures an effective improvement of these properties. The results of the tests are given in Table 27.

Table 27. Coefficient of friction (steel on steel).

Working temperature of oil °C	Oil without additive	Oil with INKhP-46 additive
20	0,09	0,05
40	0,09	0,06
60	0,09	0,05
80	0,09	0,05
100	0,09	0,05
120	0,10	0,05
140	0,10	0,05
160	0,11	0,05
180	0,12	0,05
200	0,28	0,06
220	Jamming	0,06
300	Jamming	0,06

Thus the maximum value of friction coefficient in the case of application of INKhP-46 additive is lowered by more than 5 times, and the minimum – by almost twice. Furthermore, oil without an additive at a temperature in the zone of contact of over 200° is noneffective, but with the use of an additive the limit of efficiency is increased up to 300°.

The appraisal of antiwear and antiscoring properties of oil with 3% INKhP-46 additive was done according to the method of determination of generalized index of wear based on GOST 9490-60. For oil without an additive the value of this index was 20, and for oil with 3% INKhP-46 additive – 96. Consequently the antiwear and antiscoring properties of oil with the addition of INKhP-46 additive are improved considerably.

Bench tests were conducted on reduction gears TsDN-300, TsDN-650 and TsDN-750 under conditions characterizing alternating loads. A typical curve of dependence of torque on angle of rotation of the crank is shown in Fig. 39. Industrial-45 oil with 3% INKhP-46 additive was tested on experimental samples of TsDN-650 reduction gears on rocking machines with Novikov gearing with a load which alternated in value and sign. For TsDN-650 reduction gear the nominal load subjected to check was 2500 kg-m on the driven shaft. The tests were conducted with an overload with respect to nominal of 1.6 times (4000 kg-m). Duration of the tests was 860 h. After breaking in industrial-45 oil without additive and with 3% INKhP-46 additive was poured into the washed housings of the reduction gears.

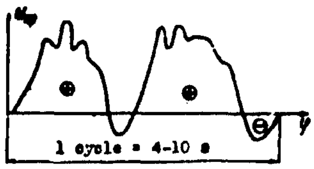


Fig. 39. Dependence of torque on the angle of rotation for the reduction gear of a rocking machine.

In the process of testing the condition of the working surfaces of the teeth was inspected with a sampling of oils in order to study the change in the physicochemical properties of scavenge oils and to determine the content of products of wear in the oil.

The onset of pitting in reduction gear No. 4 (industrial-45 oil) was fixed during the first inspection (after 150 h), and in reduction gear No. 5 (industrial-45 oil with 3% INKhP-46 additive) – during the third (380 h).

The content of deposit in scavenge oil without additive was 0.15%, and with additive – 0.11%. Viscosity of the oil changed from 7.40 to 7.80 cSt at 100° for oil without an additive and from 8.50 to 8.42 cSt with an additive. Increase of acidity for oil with an additive was 0.25 mg KOH per 100 g of oil, and for oil with 3% additive – 0.2 mg. The content of iron, determined by the

polarographic method after termination of the tests, turned out to be equal to 474.7 and 432.8 mg/kg respectively. There was practically no corrosion of scavenge oil after 251 hours of operation of the reduction gear on a lubricant with an additive, while for oil without an additive it comprised 200-300 g/m^2.

Thus:

1. Preliminary tests of the effectiveness introduction of INKhP-46 additive into industrial-45 oil showed that with 3% of this additive the antiwear and antiscoring properties of oil are improved considerably (OPI index based on GOST 9490-60 is increased from 20 for pure oil up to 96 for oil with an additive).

2. INKhP-46 additive considerably improves the antifriction properties of oil. In the range of operating temperatures from 20 to 200° the maximum value of friction coefficient when using INKhP-46 additive is reduced by more than 5 times, and minimum — almost twice.

3. INKhP-46 additive increases the temperature limit of efficiency of oil in the boundary lubricating layer.

4. Prolonged bench tests of industrial-45 oil with 3% INKhP-46 additive conducted on the TsDN-650 reduction gear under conditions of overload, showed that time of operation of the gear up to the onset of breakdowns due to contact fatigue is increased by more than 2 times.

As a result of investigation of the influence of lubrication on the longevity of toothed gears with Novikov gearing it has been established that the use of industrial-45 oil with 3% INKhP-46 additive as the lubricant of TsDN reduction gear on rocking-stands improves the basic operational qualities of the oil, due to which longevity of the gear is increased and its conditions of operation are improved (intensity of the process of crumbling due to contact

fatigue decreases, wear on gear teeth is decreased, less change is observed in the physicochemical properties of oils in process of their prolonged use in reduction gears, etc.).

On the basis of the results of tests of industrial-45 oil with 3% INKhP-46 additive in reduction gears with Novikov gearing on rocking machines it is possible to recommend it as a lubricant for these reduction gears.

3. Use of INKhP-32 Additives for Lubrication of Electric Drills Used for the Drilling of Oil Wells

In connection with continuous increase of output of oil and gas at new promising deposits in the USSR considerable attention is allotted to increasing the effectiveness of drilling, especially of deep wells. In this direction great successes have been attained with replacement of rotary drilling by a more effective drilling, when the so-called "face" motor-turbodrills and electric drills are used. However, the wide introduction of an effective method of electric drilling is restrained mainly by its insufficient reliability in operation. Improvement of it is connected in many respects with the development of the best designs of packings and an increase in the period of their service, which makes it possible to increase the time that the electric drill remains on the face.

One of the effective means for increasing the hydroprotection of electric drills is the use of additives for face packings.

In industrial equipment all greater propagation is beginning to be received by face packings [165-168].

A schematic diagram of face packing is shown in Fig. 40. A rotating shaft separates the packed working cavity from the environment by means of the simplest kinematic pair, formed by mobile and fixed rings which are connected with the shaft and body by elastic connecting elements.

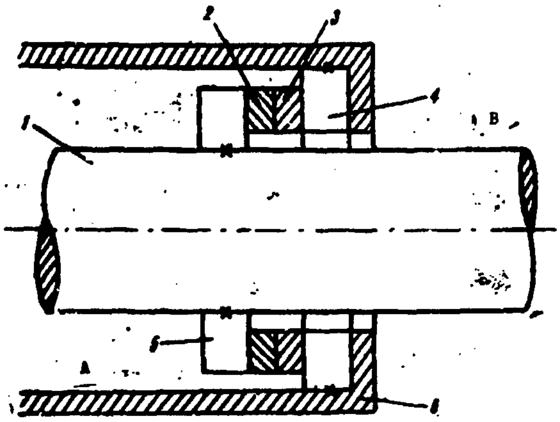


Fig. 40. Constructional diagram diagram of a face packing. A) sealed working cavity; B) surrounding area; 1 – rotating shaft; 2 – mobile ring; 3 – fixed ring; 4 – elastic coupling elements; 5 – fixed body; [Translator's note: Item #6 is not given in the key to the figure].

Numerous varieties of face packings are described in detail in the literature [169, 170, 171]. Of these we will examine the double face packings, in which various mineral and synthetic oils are used in the form of locking liquids. A change of their basic physicochemical characteristics with the help introduction of alloying additives has an influence on the conditions of operation of such a type of face packings.

As is known [172], in an analysis of conditions of operation of a lubricant in clearances between surfaces with microroughnesses consideration is given to the transmission of force between them both through adjoining crests of microroughnesses and also through the working fluid.

Operating conditions for a lubricant in clearances between contacting surfaces of face packings are examined as quasi-hydrodynamic [172, 173], taking into account the special properties of boundary lubricating layers, which, based on the definition of A. S. Akhmatov [174], make it possible to consider them as quasi-rigid. As investigations show [175, 1"6], here an anomaly of dependence of frictional force relative Newton or Bingham flow is observed.

Under such conditions the presence in a lubricant, mainly in its boundary layers, of active components, which will strengthen the effects described above or others [177], will considerably change the conditions of lubrication.

Here one should also note the essential influence of effects of chemical modification of the surface, connected with the presence in boundary lubricating layers, in the case of additives in the lubricant, of active alloying components (sulfur-, chlorine-, and phosphorus-containing compounds), which under specific temperature conditions, determined by an instantaneous increase of temperatures [70] on discrete contact, will form eutectic layers with low shifting characteristics.

Also considerable is the influence of alloying components in lubricating compositions for face packings and the ensuring of normal running in, especially in the e when there are severe conditions of friction (pressure in tens of atmospherees, velocity of tens of meters a second at a temperature of an order of 100°).

One of the most important results of the active influence of alloying components in a lubricant on the operating conditions for a face packing is a lowering of heat generation during friction due to a lowering of work of friction, and also a decrease of losses of power besides. The latter can be determined in the following way

$$N_{\text{тр}} = \frac{f}{102} \frac{\pi (D_2^2 - D_1^2)}{4} p_{\text{уд}} \frac{(D_2 - D_1)}{4} \cdot \frac{\pi n}{30} =$$
$$= 2{,}02 \cdot 10^{-4} (D_2 + D_1)(D_2 - D_1)^2 f p_{\text{уд}} n \ (\text{kW}),$$

where f – coefficient of friction, n – number of turns, D_1, D_2 – diameters of face surfaces (m), $P_{\text{уд}}$ – specific pressure (kg/m²).

In examining the linear heat flow between faces of rubbing cylinders, V. S. Shchedrov [55] obtained an equation for the stabilized process of thermal conductivity:

$$\lambda' = \frac{d^2\vartheta}{dx^2} + q = 0,$$

where λ' – coefficient of thermal conductivity of boundary lubricating layer; ϑ– temperature; q – quantity of heat, generated by a unit of volume in a unit of time.

A particular solution of this differential equation is presented in the following form:

$$\vartheta = -\frac{qx^2}{2\lambda'} + \vartheta_{max}.$$

For a case of heat generation during friction of surfaces of face packings covered by boundary lubricating films it is possible to write the equations:

$$\vartheta_1 = \vartheta_{max} - \frac{qh_1^2}{2\lambda'},$$

$$\vartheta_2 = \vartheta_{max} - \frac{qh_2^2}{2\lambda'},$$

where ϑ_1– temperature on surface of fixed bronze ring, ϑ_2– temperature on surface of mobile steel ring, λ_1 and λ_2 – respectively the coefficients of thermal conductivity of rubbing face surfaces, h_1 and h_2 – respectively the thicknesses of boundary films, λ' – coefficient of thermal conductivity of boundary layer of lubrication.

Thus by influencing on the one hand thermogeneration during friction, and on the other – the physical and thermophysical parameters of the lubricating layer on rubbing surfaces of face packings, it is possible to improve their conditions of operation.

The influence of rheological properties of lubricating layers, working in face packing, can be analyzed if there are solutions of the equation of fluid motion relative to the examined case. For a case when the temperature coefficient of viscosity is accepted as constant, a solution is given by S. P. Livshits [178], and in the case of a calculation of a variable value of this coefficient the problem has been solved by A. I. Golubev [169]. The influence of viscosity on distribution of pressure in a clearance is examined in work [179].

There is special interest in the so-called "Reyner effect" which is described in the literature [180]. It entails the appearance of a pressure drop from the periphery to the center. If this drop overcomes the influence of forces of inertia, then the flow of liquid will occur from the periphery of the pair to the center. As a possible cause of the appearance of this drop reference is made to the axial oscillations of the rotating ring of the pair, the frequency of which is equal to the doubled angular frequency of rotation of the shaft.

For the case of operation of the double face packings of electric drills the Reyner effect acquires specific importance. If the packing separates the working fluid, surrounding the friction pair along the outer diameter, from the locking liquid, and the pressure of the locking liquid differs little from the pressure of the working fluid, then under pressure, conditioned by a drop due to the Reyner effect, the entry of working fluid into the locking fluid is possible.

The influence of alloying components in a lubricant on the efficiency of a face packing is also examined by us proceeding from the fact that the product of average specific pressure on rate of sliding $p_{y\mu}\cdot v$ is taken as the determining parameter. This parameter is connected directly with the lifetime of the packing. Thus at $pv = 60$, according to certain firms, a lifetime of around 2000 hours is ensured.

The intensity of wear of face pairs at rated values of average specific pressure and speed of slip is reduced sharply with the addition of antiwear additives to the locking liquid. Antiscoring additives prevent local gripping, jamming, and scores, a result of which is usually breakdown of the face packing. Antiwear and antiscore effects of the action of additives make it possible to design face packings with high values of the parameter pv.

Investigation of antiscoring and antiwear properties of alloying locking liquids by means of determination of generalized wear index (OPI) [181] on a four-ball friction machine showed that the OPI value with the introduction of additives increases from 20-25 for ordinary mineral oils to 100-110.

R. M. Matveyevskiy [110] established that for every mineral oil there is a dependence

$$P_{max} v \mu = C$$

where P_{max} – maximum load, v – rate of slip, μ – coefficient of friction, C – constant.

With the addition of antiscoring additives to oil the value of C due to an increase of P_{max} can be increased by 1.5-2 times.

An increase of maximum load capacity of a lubricating layer is usually accompanied by a lowering of wear of rubbing surfaces.

With the addition of additives it is possible to improve other operational characteristics of working fluids – anticorrosion, antioxidant, demulsifying, and others.

Proceeding from this an investigation was made of the possibility of application of additives for increasing the reliability of hydroprotection of electric drills, in which face packings are used extensively.

Contemporary electric drills consist of an electric motor and spindle, most frequently oil-filled. Sealing of both ends of the shaft of the motor is achieved with the help of lower and upper collar seals with lubricators. Furthermore, the coupling of shafts of the oil-filled spindle and the motor shaft is hermetically sealed by hinged packing, and the outlet lower end of the spindle shaft – by a collar.

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Conditions of Work of Face Packings in Electric Drills and Requirements Presented for the Quality of Lubricant

Face packings in electric drills work under conditions of lubrication with high-viscosity residual MK-22 oil which is prepared from high-quality oils from the apsheronsk deposits – Surakhansk and Karachukhursk. Oils obtained on a base of these petroleums possesses high operational qualities – their high degree of purity ensures good indices of viscosity-temperature properties and high stability. For a determination of basic requirements which should be satisfied by oil for packings which are used in electric drills we will examine their conditions of work.

System of protection of electric drills. This system anticipates the filling of the inner cavity of the electric drill with oil and the development of counterpressure, equal or greater than the pressure of the medium surrounding the electric drill. This counterpressure is created by a special system of lubricators, which transmit external pressure into the oil cavities by means of pistons moving in a cylinder. An additional pressure on the piston is the action of a spring. The excess of internal pressure over pressure comprises 1-3 kg/cm^2.

Such a balancing action of the lubricator pistons remains constant under any external conditions (depth of submersion, temperature on the face, and others).

In electric drill with a spindle on antifriction bearings the entire electric drill is filled with oil; the motor with insulating and the spindle with lubricating. The motor and spindle are separated by a collar and have independent lubricators. The reliability of packings, in which any constructions of face collars are used, depends on the value and direction of drops in pressure which is effective in places of packings. Therefore for characterizing the conditions of work of packings it is necessary to know the values of pressure drops on separate sections of the electric drills. They depend on quantity and properties of the washing fluid and the direction of its movement.

Pressure drops is classified as in a state of rest, during direct washing, and during reverse washing. Values of pressure drops in the examined cases are determined in the following way [162].

<u>Pressure drop in a state of rest</u>. The oil filling the inner cavity of the electric drill is under excess pressure, created by the spring of the lubricators.

The value of this pressure p: $p_{min} < p < p_{max}$,

where

$$p_{max} = \frac{W_{n\,max}}{F},$$

$$p_{min} = \frac{W_{n\,min}}{F},$$

$W_{n\,max}$ – maximum force of spring in upper limiting position, $W_{n\,min}$ – minimum force of spring in lower limiting position, F – area of piston.

Due to the difference of specific gravities of the washing fluid $\gamma_{ж}$ and the oil $\gamma_{м}$ the pressure drop, created by the lubricator spring, decreases by a value equal to

$$\Delta p_{\gamma} = h\,(\gamma_{ж} - \gamma_{м}),$$

where h – height of oil column.

Here it is natural that

$$p_{ж.ш} > \Delta p_{\gamma}.$$

<u>Pressure drop during direct washing</u>. Losses of pressure ΣH are determined by the sum

$$\Sigma H = \frac{\Delta p}{\gamma_ж} = \frac{\Delta p_{в.дв.} + \Delta p_{в.ш.} + \Delta p_{дол.}}{\gamma_ж},$$

where $\frac{\Delta p_{в.дв.}}{\gamma_ж}$ — loss of pressure during circulation of liquid through shaft of the motor, $\frac{\Delta p_{в.ш.}}{\gamma_ж}$ — the same, through shaft of spindle, $\frac{\Delta p_{дол.}}{\gamma_ж}$ — the same, through the bit. For ordinary bits Δp = 3-20 kg/cm^2, for monitor — $\Delta p \approx 80$ kg/cm^2.

Maximum pressure, obtainable during the operation of monitor bits, considerably exceeds normal pressures at which reliable hydrodynamic lubrication is ensured, and therefore it is natural the danger of scoring and jamming arises on those rubbing surfaces with which this pressure is transmitted.

<u>Pressure drop during reverse washing</u>. With this washing there is a change in the location of danger zones where the maximum value of pressure drop can appear. A special case of reverse circulation of washing fluid is its movement during descent of the electric drill onto the face. In this case, as calculations show, an additional pressure of up to 5 atm appears.

The following factor, determining the condition of work of the packing, are speed parameters, which depend on type of electric drills used. Table 28 gives the data on basic types of electric drills used in Soviet industry.

Table 28. Basic data on electric drills

Indices	Code				
	Э170/6	Э215/10	Э215/8	Э250/10	Э250/8
Rate of rotation of shaft, r/min	1000	600	750	600	750
Power of motor, kW	100	120	150	150	230
Voltage, V	1000	1100	1210	1100	1650
Outer diameter, mm	170	215	215	250	250
Diameter in places of threaded connections of bodies of electric drill, mm	173	220	220	255	255
Length, m	11,5	12,5	12,5	12	13,2

Increasing the Reliability of Face Packings of Revolving Shafts of Electric Drills by the Application of Additives

As was already indicated, the packings of revolving shafts of electric drills determine the period of service of the winding of the electric motor and, consequently, the whole unit. Packings work under severe conditions which are determined by a number of factors: vibration of shafts, play and sagging of shafts, action of abrasive particles of the washing fluid on packing surfaces, high hydrostatic pressure, variable pressure drop, high temperature.

An analysis of causes of breakdown of electric drills shows that this occurs most frequently from malfunctioning of the upper collar. Thus 46% of malfunctions of electric drills is combined with disruption of work of the upper subassembly for hydroprotection. Also a result of disturbance of normal function of the hydroprotection is breakdown of the motor due to breakdown of the winding (around 25%). Causes connected with the work of the spindle collar make up 9.1% of the cases.

The addition of hematite in the washing fluid and the appearance of vibrations with an increase of mechanical strength of drilled rocks cause a lowering of stability of the hydroshielding of electric drills.

The influence of the abrasive medium increases with a lowering of viscosity of the washing solutions. In connection with the predominance of sandy rocks the abrasive action of the flushing solution is increased in discontinuity formations, NGK, and NKP. When sinking through these formations sharp lowering of the interspection periods of operation of electric drills is observed, and when sinking through formations of KS and PK, where clay layers in the rocks, predominate this period increases.*

<u>Principle of operation of face packings in electric drills</u>. The face packing (Fig. 41) of an electric drill carries out its functions due to the creation of a great deal of hydraulic resistance between the ends of the two rings, one of which is fixed, mounted in the body, and the other revolves, being pressed to the first by springs and pressure drop.

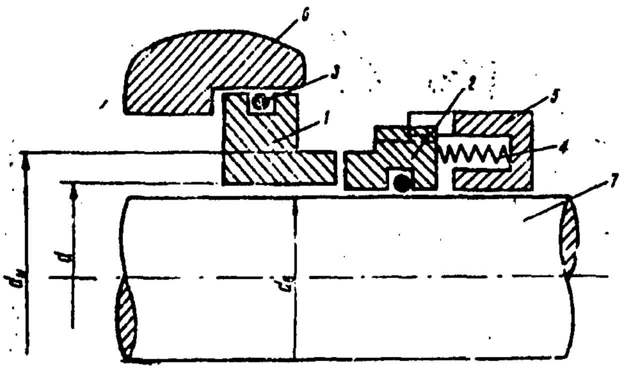

Fig. 41. Diagram of face packing for an electric drill. 1 – fixed ring; 2 – steel ring; 3 – rubber packers; 4 – spring; 5 – bushing; 6 – housing; 7 – shaft.

*[Translator's note: The following acronyms were used in relation to geological formations: KS – kirmakinskaya formation; PK – podkirmakinskaya formation; NKP – nadkirmakinskaya sandy formation; NGK – unidentified].

The face collar consists of fixed ring, made from antifriction bronze and mounted in the housing, and a mobile steel ring, revolving together with the shaft and pressed by its end to the end of the fixed ring with the help of springs.

During the operation of such a type of face packing conditions can arise when the regimen of purely fluid lubrication is disturbed and friction passes into boundary conditions. High pressure drops, the presence of an abrasive, and high ambient temperatures create favorable conditions for disrupting the hydrodynamic conditions of friction. Under these conditions the lubricating oil, the effectiveness of which was determined mainly by the value of coefficient of internal friction (viscosity), should also possess a number of other properties to ensure the reliable operation of this face packing under conditions of boundary friction. The most important properties of lubricating oils under such conditions, as is known [149, 183-191], are antiscoring, antiwear and anticorrosion.

As numerous investigations show [192-195], these properties in the initial mineral raw material, which is to be treated for the purpose of obtaining high-quality lubricating oils, appear very insignificantly. Meanwhile, they can be sharply strengthened by the addition to the oil of special alloying additions — additives.

The use of antiscoring additives makes it possible to increase the carrying capacity of the boundary layer of lubrication under conditions of friction, when conditions are not ensured which are necessary for preservation of sufficient hydrodynamic "wedge." Under conditions which are characteristic for the work of face packings this will signify that at high drops, when the danger of onset of boundary friction appears, a reliable and durable protective film will appear on the face surfaces.

Antiscoring additives also make it possible to increase specific loads on adjoining surfaces of the rings of face packings, i.e., the danger of transition from conditions of liquid lubrication to boundary can be shifted to the area of large pressures.

Antiwear and anticorrosion properties of such additives are manifested mainly in a lowering of abrasive and corrosion wear, the danger which will take place in case of application of a clay solution as the washing fluid.

Thus, summarizing what was said, it is possible in the following way to determine main requirements which are presented for the quality of a lubricant for face packings of electric drills by the conditions under which they work:

1. Oil should possess as high as possible a level of viscosity and high viscosity-temperature properties, which will ensure reliable lubrication at the high temperatures which occur on the face.

2. Oil should possess high antiscoring properties, i.e., ensure sufficient strength of the oil film under conditions of boundary friction and prevent gripping and jamming of working surfaces of face pairs, which can lead to emergency breakdowns. This will make it possible to move into the area of higher pressures the maximum value of drop, at which the danger of transition of lubricating conditions from hydrodynamic to boundary arises.

3. Oil should ensure minimum wear under conditions of abrasive action of particles of the drilling mud and bring to a minimum its possible corrosive action.

In accordance with these requirements we conducted selection tests of a number of additives. In particular, the synthesized additive INKhP-32 constitutes a sulfur- and phosphorus-containing compound, the addition of which to lubricating oil lowers friction

and wear creates a durable protective film on rubbing surfaces, and prevents corrosion of surfaces.

Additive INKhP-32 is obtained by means of condensation of alkylphenol with formaldehyde in an alkaline medium in the presence of a solvent (n-butyl alcohol) and a 10% solution [Translator's note: At this point a portion of text has apparently been left out] is loaded 1 mole of formaldehyde (37% solution of formaldehyde in water). As much solvent (n-butyl alcohol) and 10% solution of NaOH is added to this mixture, the mixture is thoroughly mixed, and then it is left for 30 hours without mixing. Then the reaction mixture is neutralized with hydrochloric acid and diluted with benzene. It is washed to a neutral reaction. The benzene and n-butyl alcohol are distilled. The condensed product is treated with 20% P_2S_5, is filtered, and treated anew with barium hydroxide. The resulting additive contains 2-2.5% phosphorus, 14.8-15%, ashes and 4-4.5% sulfur.

For obtaining the lubricating composition 6% by weight of additive is added to the oil.

Certain physicochemical characteristics of the composition obtained are given in Table 29.

Table 29. Physicochemical characteristics of the composition.

Indices	MK-22 oil	
	Without additive	with 6% INKhP-32 additive
Specific weight	0,9030	0,9050
Kinematic viscosity (cSt) at		
100°	22,8	24,9
50°	186	210
Flash point, °C	283	247
Content of ashes, %	0,0065	0,133

The resulting composition of oil with additive passed preliminary selection tests on special installations, and also industrial tests under commercial conditions.

Preliminary tests were conducted on a four-ball friction machine (ChShM) by a special method. The tests consisted of measurement of the diameter of the spot of wear on each of three fixed balls, which with a fourth revolving ball make up the so-called "four-ball pyramid." Friction was carried out at 1400 r/min on the upper ball of the pyramid with an axial load of P = 27.6 kg, which with diameter of balls of 19 mm corresponded to an initial contact pressure based on Hertz of σ = 16,000 kg/cm^2. Duration of the test was 6 hours. Measurements of friction track on the fixed balls were made after 15 min, 2, 4, and 6 hours of friction under the assigned conditions of loads and ball of slip.

Various combinations of known additives were tested in different concentrations of added material in the same initial oil. The results of the tests are given in Table 30.

Table 30. Results of tests on a ChShM.

Oil	Average wear (mm) after work			
	15 min	2 h	4 h	6 h
Initial (without additive)	0,62	0,90	1,20	1,38
The same + INKhP-30 additive				
1%	0,58	0,84	1,09	1,18
3%	0,53	0,80	1,04	1,09
5%	0,51	0,79	1,01	1,21
The same + 5% INKhP-31A additive	0,49	0,72	0,82	1,02
The same + 5% INKhP-31 additive	0,60	1,28	1,53	1,64
The same + INKhP-32 additive				
1%	0,62	1,21	1,92	2,00
3%	0,60	0,93	1,11	1,48
6%	0,45	0,54	0,58	0,61
7%	0,43	0,53	0,63	0,70

As can be seen from these data, the greatest effect is obtained with the addition of 6% INKhP-32 additive, which was accepted for further bench tests. An increase of its concentration to 7% shows even a certain tendency to impairment of effectiveness.

We will give the results of bench tests, conducted under conditions simulating the work of face packing for collars in electric drills.

For the test we prepared two samples – MK-22 oil without an additive, used for filling lubricators in the upper collars, and the same oil with the addition of 6% INKhP-32 additive. The tests were conducted on a LTTO machine (Leningrad Polytechnical Institute), in which load on fixed bronze blocks is carried out up to a load corresponding to the beginning of jamming.

The central steel rod of the LTTO friction machine, which is pressed by two fixed bronze blocks, can revolve with a various number of turns (6 variants), which with a rod diameter of 18 mm correspond to the values of rates of slip relative to the fixed bronze blocks in 0.115; 0.243; 0.497; 0.900; 1.050; 3.77 m/s.

In the selection of speed conditions for testing the antiscoring properties of oil additives the values of peripheral velocities on face surfaces of collars for Soviet models of electric drills were accepted as initial.

The test oil was poured into the circulation system, was heated up to the temperature at which the test was carried out, and after running in of the bronze blocks to the steel rod, which was rotating at a speed corresponding to the rate of slip on face pairs, gradual loading of the arms by the spring loader was begun. When there was a sharp jump in friction moment, indicating gripping and jamming of the fixed bronze blocks which were rubbing against the revolving steel rod, the experiment was ceased. Load at beginning of jamming was taken as the appraisal index.

The tests showed that the maximum pressure, at which jamming of a steel-bronze pair occurs ($V = 3.77$ m/s), in the case of working on MK-22 oil without an additive comprises 41.5, and with 6% INKhP-32 additive – 77.5 kg/cm^2.

Thus the addition of 6% INKhP-32 additive to MK-22 oil increases its maximum load capacity by more than 1.5 times.

The MK-22 oil with 6% INKhP-32 additive was also tested under commercial conditions at the experimental drilling office of the "Azneft'" association. In various periods the test lubricant was poured into electric drills, working in parallel with electric drills on a regular lubricant. Table 31 gives comparative data on the performance of electric drills with the test and ordinary lubricants in a depth interval of 2000-2400 m, where the greatest number of electric drills worked.

Table 31. Comparative data on the working of electric drills.

Test lubricant						Ordinary lubricant					
No. of bore-hole	No. of electric drill	Interval of drilling, m		Sinking, m	Time of drilling, h	No. of bore-hole	No. of electric drill	Interval of drilling, m		Sinking, m	Time of drilling, h
		from	to					from	to		
278	062	2062	2082	20	4,5	97	329	2005	2037	32	4,0
97	59	2151	2188	37	7,25	165	2030	2093	2149	56	13,75
915	2033	1909	2006	97	19,0	165	16	2050	2093	43	5,5
278	20	2257	2309	52	16,5	97	27	2037	2063	26	2,5
97	062	2213	2271	58	11,75	278	59	2000	2062	62	20,0
						165	2027	2149	2160	11	3,0
279	2027	2184	2257	73	34,5	165	062	2160	2274	114	16,5
165	033	2317	2400	83	26,5	97	329	2063	2157	85	16,75
915	2030	2155	2215	56	20,0	97	329	2188	2213	25	3,5
						165	31	2400	2410	10	6,5
278	019	2309	2448	139	31,75	97	329	2307	2337	30	9,5
On the average for one electric drill				68,4	19,4					45,2	9,13

As one can see, when working on the test lubricant sinking was increased by one and a half times, and the interinspection period – by more than 2 times. In a visual inspection of the surfaces of face pairs after working on the test oil their rubbing surfaces in a satisfactory condition. A darkening of the surfaces of the face pairs was observed. This was apparently caused by the action of the active part of the additive. Traces of corrosion and intense wear on the surfaces of pairs were not observed. Rubber packings after working of face collars on test oil were in a satisfactory state. Expenditure of viscous oil with using the test lubricant turns out to be less than with an ordinary lubricant.

Thus an increase of work effectiveness of electric drills can be atained by improvement of conditions of work of the face packings by means of adding to the lubricating oil antiscoring, antiwear or anticorrosion, and also multifunctional additives. Of the tested additives INKhP-30, INKhP-31, and INKhP-32 the best antiwear properties are possessed by INKhP-32 additive at an optimum concentration of 6%.

Bench test of a mixture of 6% INKhP-32 additive with MK-22 oil, which were conducted on an LTTO friction machine by the method of the INKhP of the Azerbaydzhan SSSR Academy of Sciences under conditions reproducing the work of face surfaces of collars for electric drills, showed an increase in the load capacity of oil by more than 1.5 times.

In industrial tests of electric drills with the test lubricant (MK-22 with 6% INKhP-32 additive) an increase was achieved in the period of service of face packings and a lowering of expenditure of oil. Conditions are improved for the exploitation of electric drills due to an increase in the reliability of work of face packings (bronze-steel).

It is necessary to consider that antiwear, antiscoring, and anticorrosion properties of the test additive are the basic factors, promoting an improvement in the operational qualities of the previously used MK-22 oil.

Improvement of the qualities of lubricants for collar packings of electric drills by the introduction of effective additives is not only a promising means of increasing the reliability of electric drills, but also opens the path for a number of new design solutions.

Additional tests were also conducted in which in the same electric drills ordinary and test lubricants were used alternately. Thus the influence of almost all outside factors, connected with conditions of assembly and different assembly clearances in couplings, was reduced to a minimum.

Electric drills with the test lubricant penetrated ranges of depths from 2101 to 2809 m, and with ordinary – from 2230 to 3190 m. Table 32 gives the average data based on the results of tests of the test lubricant on E-215/8 electric drills.

Table 32. Average data based on the results of tests of a test lubricant as the sealing fluid for face packings of E-215/8 electric drills.

Appraisal indices	MK-22 without additive	MK-22 with 6% INKhP-32 additive
Extent of sinking for one conditional electric drill, m	24.5	45.5
Time of drilling, h	9.5	17.5
Wear of upper collar, mm		
Rings of lower pair		
bronze	0.4	0
steel	0.2	0.1
Rings of upper pair		
bronze	0.05	0.15
steel	0.2	0.15
Wear of lower collar, mm		
Rings of lower pair		
bronze	0.05	0.06
steel	0.05	0.06
Rings of upper pair		
bronze	0.1	0.07
steel	0.1	0.15

As can be seen from data in the table, average value of sinking for one conditional electric drill and time of drilling were increased by 1.8 times. The data obtained are in sufficiently close correspondence with indices of tests conducted earlier. For example, based on the results of previously conducted tests the average time of drilling had values of 9.1 and 19.4 hours respectively for oil without additive and with an additive, and according to the results of the tests described – 9.5 and 17 hours respectively.

Thus it is possible with sufficient confidence to consider that the use of MK-22 oil with 6% INKhP-32 additive as the packing fluid for face packings of electric drills ensures an increase in their period of service.

As can be seen from data in Table 33, the test oil in process of work is less subject to a change of its initial characteristics. Thus a change of kinematic viscosity at 100° composed for ordinary oil comprised 8.3 cSt, and for test – only 2.3 cSt. An analogous picture is also observed with the increase of viscosity of scavenge oil at 50° (change of 73 and 33 cSt respectively.

Table 33. Change in the physicochemical properties of scavenge oils.

Indices	Method of testing	MK-22 oil	
		ordinary	test
Kinematic viscosity at 100°, cSt	GOST 33–53		
initial oil	"	22.8	24.9
scavenge	"	31.1	27.2
The same, at 50°, cSt	"		
initial oil		186	210
scavenge		259	243
Content of water in scavenge oil, %	GOST 2477–44	14.1	3.7
Content of mechanical impurities in scavenge oil, %	GOST 6370–52	11.4	1.2
Test of corroding action of scavenge oil on metals			
steel surface	GOST 2917–45	Does not withstand	Withstands
copper surface	—		
Content of ashes, (%) in			
initial oil	—	0.0065	0.183
scavenge	—	0.268	0.214

Reliability of work of the hydroshield is characterized to a considerable degree by the intensity of contamination of oil with impurities and it getting into the drilling mud. As can be seen from the data in the table, the test oil is less subject to irrigation less mechanical impurities get into it due to the more satisfactory contact of rubbing surfaces of face pairs of the collar packings. The improved antifriction characteristics of the lubricant undoubtedly lead to less heat generation on the friction surfaces, which essentially improves the condition of contact.

CHAPTER VI

WAYS OF STANDARDIZATION OF OILS USED FOR THE LUBRICATION OF INDUSTRIAL EQUIPMENT

Conditions of operation of the basic items of industrial equipment are monotypic in many respects. Therefore it is expedient to examine the possibility of standardization of lubricating materials used.

Two directions are possible for the standardization of the assortment of oils for the lubrication of industrial equipment: rational combining of basic sorts and selection of rational compositions of additives. Here the second path does not exclude the possibility of preliminary reduction of the assortment of the main base sorts of oils.

Standardization on the Basis of Base Oils

Based on the level of viscosity of oil used for the lubrication of industrial equipment the following ranges of kinematic viscosity are included: ν_{50} = 10-60 cSt and ν_{100} = 6-35 cSt [194].

In an analysis of the quality of commercial oils it is clear that for low-viscosity oils the same sorts (based on level of viscosity) cover the ranges of 20-23, 18-33, and 38-51 cSt (at 50°), and for oils of average and high viscosity – 10-13, 20-22, 24-28, and 28-30 cSt (at 100°).

It is natural that besides the indices of level of viscosity one ...ould consider the differences in other qualitative characteristics of oils used. For example, in the range viscosity of 17-23 cSt (at 50°) three types of oils are used – axial 3, turbine Л, and industrial-20 (spindle-3). A comparison of their basic physicochemical indices is given in Table 34.

Table 34. Comparison of basic characteristics of oils with a kinematic viscosity within the limits from 17 to 25 cSt (at 50°).

Indices	Axial-3	Turbine-22 (Л) based on GOST 38-53	Industrial-20, spindles-3
Viscosity, kinematic at 50°C, in cSt	20–25	20–23	17–23
Acid number (mg KOH per 1 g of oil) no more than	—	0,02	0,14
Ash content (%), no more than	—	0,005	0,007
Content of water-soluble acids and alkalis	Absent	Absent	Absent
Content of mechanical impurities	-	Absent	Absent
Flash point in open crucible (C), lower than	130	180	170
Pour point (°C), no more than	-40	—15	—20
Sodium test based on GOST 6793-53 with acidification (points) no more than	—	2	—
Transparency at 0° C	--	Transparent	--
Water (%), no more than	0,3	—	—
Stability based on GOST 981-55			
deposit after oxidation (%), no more than	—	0,1	-
acid number after oxidation (mg KOH per 1 g of oil), no more than	—	0,35	—
Rate of demulsification based on GOST 1321-51 (min) no more than	—	8	—

Axial 3, which is light in fractional composition, has a minimum pour point (-40°). Obtaining of an oil corresponding in characteristics to axial 3 can be carried out by means of compounding heavy and light components using the effect of eutectic lowering of the pour point.

A comparison of the characteristics of turbine-22 (turbine Л) and industrial-20 permits the assumption that bringing the qualities of the latter up to the level of turbine can be ensured only by the application of antioxidant and anticorrosion additives.

The same can be said in comparing the indices of industrial-30 (machine Л) and turbine-30 (see Table 35).

Table 35. Comparison of the basic characteristics of oils with a kinematic viscosity within the limits from 27 to 33 cSt (at 50°).

Indices	Industrial-30 (machine Л)	Turbine-30 (Ut) (GOST 32-53)
Viscosity, kinematic 50°C, in (cSt)	27—33	28—32
Acid number (mg KOH per 1 g of oil), no more than	0,2	0,02
Ash content (%), no more than	0,007	0,005
Content of water-soluble acids and alkalis	Absent	Absent
Content of mechanical impurities	0,007	Absent
Flash point in open crucible (°C), no less than	180	180
Pour point (°C), no more than	—15	—10
Sodium test based on GOST 6473-53 with acidification (points), no more than	—	2
Transparency at 0°C	—	Transparent
Stability based on GOST 981-55		
deposit after oxidation (%), no more than	—	0,10
acid number after oxidation (mg KOH per 1 g of oil), no more than	—	0,35
Rate demulsification based on GOST 1321-51 (min), no more than	—	8

The following group of oils, with a level of viscosity of an order of 40-60 cSt at 50°, is characterized by the quality indices given in Table 36. This group is also composed of three basic sorts of oils: industrial, turbine, and axial. Bringing the indices of industrial oils up to the level of turbine can also be ensured here only by the addition of effective antioxidant and anticorrosion additives.

In examining the given characteristics of low-vicosity sorts of oils (10-60 cSt at 50°), the conclusions can be made that standardization of monotypic samples of turbine and industrial oils is possible by the selection of sufficiently effective antioxidant, anticorrosion, and demulsifying additives, or by improvement of the technology of obtaining industrial oils.

For oils of average and high level of viscosity a still greater assortment is characteristic. A comparison of the basic physicochemical characteristics of two large groups of oils – of a level of around 10 cSt at 100° and within limits of 15-30 cSt with that same temperature – is given in Tables 37 and 38.

Table 36. Comparison of basic characteristics of oils with a kinematic viscosity within the limits from 36 to 59 cSt (at 50°).

Indices	Industrial 45(S)	Industrial 50 (CU)	Turbine-46 (T) based on GOST 32-53	Axial Л	Turbine-57 (turboreduction gear) based on GOST 32-8
Viscosity, kinematic at 50°C (cSt) within limits of	38–52	42–58	44–48	36–52	55–59
Acid number (mg KOH per 1 g of oil), no more than	0,35	0,15	0,02	—	0,05
Ash content (%), no more than	0,007	0,005	0,020	—	0,040
Content of water-soluble acids and alkalis	Absent	Absent	Absent	Absent	Absent
Content of mechanical impurities	0,007	0,007	Absent	0,07	Absent
Flash point in open crucible (°C), not below	190	200	190	135	195
Pour point (°C), no more than	—10	—20	—10	—15	—
Sodium test based on GOST 6473-53 (points) no more than	—	—	2	—	2
Transparency at 0°C	—	—	Transparent	—	Transparent
Water (%), no more than	—	—	—	0,4	—
Stability based on GOST 981-55					
deposit after oxidation (%), no more than	—	—	0,15	—	—
acid number after oxidation (mg KOH per 1 g of oil), no more than	—	—	0,45	—	—
Rate of demulsification based on GOST 1321-51 (min), no more than	—	—	8	—	8
Coking capacity (%), no more than	0,3	0,2	—	—	—

In the first group of oils it is natural to be oriented on cylinder-11, inasmuch as application of motor oil $AK_{п}$-10 for the lubrication of industrial equipment, in spite of its considerable resources, is inexpedient due to the practical ineffectiveness of multifunctional additives found in its composition under conditions of operation of the majority of types of machines and mechanisms of industrial equipment.

Table 37. Comparison of basic characteristics of oils with a kinematic viscosity within the limits from 9 to 13 cSt (at 100°).

Indices	AK_n-10 (GOST 1862-50)	Cylinder il (cyl.-2) GOST 1841-51
Viscosity, kinematic (cSt), at 100°, no more than	10,0	9—13
Ratio of kinematic viscosity at 50° and 100°, no more than	7,0	—
Coking capacity (%), no more than*	0,40	0,8
Acid number (mg KOH per 1 g of oil) no more than	0,15	0,3
Ash content (%), no more than	0,015	0,03
Content of water-soluble acids and alkalis	Absent	Absent
Content of mechanical impurities, %	Absent	No more than 0.007
Content of water	Traces	Absent
Flash point in open crucible (°C), no more than	200	215
Pour point (°C), no higher than	—25	+5
Corrosion based on GOST 3788 (g/m^2), no more than	10	—
Color of oil (mm), no less than	4	—

*For oil without additive.

There is considerable interest in oils of the second group – with a range of viscosity of 15-30 cSt (Table 38). Nigrols, produced by industry according to GOST 542-50, constitute the unpurified residual product (mazut) from the distillation of Balakhansk heavy oil. Lightened maxut, obtained by mixing with solar oil fractions, are the basis of oils TA_n-10 with TA_n-15, to which additives Lz 6/9 and others for the improvement of antiscoring and antiwear properties. These oils, although not used widely for the lubrication of industrial equipment, however, just as thickened oils, are of interest from the point of view of the possibility of controlling the level of viscosity of standardized oils. Comparative characteristics of viscosity-temperature properties of certain oils with a level of viscosity up to 30 cSt at 100°, obtained from a promising mixture of Apsheronsk oils, are given in Table 39; qualitative characteristics of compounds, purified, and thickened oils obtained in this manner – Table 40. Samples of base oils were obtained in the laboratory of oil technology at the INKhP of the Azerbaydzhan SSR Academy of Sciences under the leadership of R. Sh. Kuliyev.

Table 38. Comparison of basic characteristics of oils with a kinematic viscosity within the limits from 15 to 30 cSt (at 100°).

Indices	AK-15 (GOST [illegible])	Nigrol Л	Nigrol З	Cylinder-24 (vis-[illegible]cosine) GOST 1341-51	P-28 (GOST .480-53)
Viscosity, kinematic, (cSt) at 100°	No less than 15	28,4–32	18–22	20–28	26–30
Ratio of kinematic viscosity at 50° and 100°, no more than	9,0	—	—	—	—
Coking capacity (%), no more than	0,70	—	—	2,5	1 0
Acid number (mg KOH per 1 g of oil), no more than	0,20	—	—	—	0,1
Ash content (%), no more than	0,015	—	—	0,05	—
Content of water-soluble acids and alkalis	Absent	Absent	Absent	Absent	Absent
Content of mechanical impurities, %	Absent	No more than 0,06	0,05	0,1	Absent
Content of water, %	Traces	Traces	—	No more than 0,05	Absent
Flash point in open crucible (°C) not below	220	180	170	240	285
Pour point (°C) no higher than	–5	–5	–20	—	–10
Color of oil, no less than	2	—	—	—	—
Corrosion of steel and copper plates, based on GOST 2417-45 at a temperature of 100° for 3 h	—	Withstands	Withstands	—	Withstands

Table 39. Viscosity-temperature characteristics of transmission oils obtained from oils from the Apsheronsk deposits.

Samples of oil	Kinematic viscosity (cSt) at		Dynamic viscosity (poise) at			Index of viscosity
	100°	50°	0°	–5°	–10°	
Cylinder-6 + spindle-2 (87% + 13%)	29,7	368	768	1841	—	36
The same, (77% + 23%)	20,7	187,1	271	506	—	60
AK-15	20,6	219,6	190	434	1129	25
Cylinder-24	21,1	189,9	230	481	1187	52
Distillate Balakhansk heavy oil with acid-contact purification	10,1	75,8	49	249	1566	—
The same, thickened with 1.2% polyisobutylene	16,3	126,7	69		304	61
Mixture of mazut from Balakhansk heavy oil and solar	20,0	207,7	221	495	1274	—
Nigrol based on GOST 542-50	30,9	335	833	1287	—	41

Table 40. Results of preliminary tests of oils obtained from oils from the Apsheronsk deposits.

Sample of oil	Corrosion of steel plates in moisture (according to Pinkevich method), g/m²	Stability during oxidation (AK-2)				OI based on GOST 9490-60
		Viscosity at 100°C, cSt		Increase, cSt	Deposit, %	
		prior to oxidation	after oxidation			
Cylinder-6 + spindle-2 (87% + 13%)	0,58	29,7	38,6	8,9	1,1	24,9
The same, (77% + 23%)	0,12	20,7	28,5	7,8	3,93	26,4
AK-15	0,44	20,6	29,1	8,5	4,10	27,3
Cylinder-24	0,56	21,1	24,7	3,6	2,12	17,1
Distillate of Balakhansk heavy oil with acid-contact purification	0,46	10,1	11,9	1,8	1,29	24,4
The same, thickened with 1.2% polyisobutylene	0,84	16,3	17,6	1,3	1,02	23,5
Mixture of mazut of Balakhansk heavy oil and solar	0	20,0	26,7	6,7	3,56	28,8
Nigrol based on GOST 542-50	0,5	30,9	35,0	4,1	2,67	32,4

On the base of these compounds and purified oils of the AK-15 type it is possible to carry out the replacement of low-quality nigrol.

Investigations showed that with the use of purified transmission oils for the lubrication of gears it is possible to attain a sharp increase in the period of service of gears, since the wear of working profile surfaces is reduced and fatigue strength of surface layers is increased. This makes it possible to reduce the harmful phenomena of pitting. Furthermore, purified oils are stabler. For example, replacement of summer nigrol by avtol-18 (AK-15 based on GOST 1862-51) ensures a considerably greater period of service of toothed gears. Table 41 gives certain data from comparative tests of AK-15 (avtol-18) and nigrol. Analogous results were obtained during tests of transmission oils with a lower level of viscosity.

A mixture of residual and distillate components of oils from sulfurous oils of selective purification (TS-14.5 based on VTU [departmental technical specifications] 110-61) eliminates the pitting which takes place during tests under analogous conditions

of commercial nigrol based on GOST 542-50. The use of purified oil: of the type AK-15 (avtol-18) instead of nigrol also ensures a high operational reliability of the equipment and long periods of service of oil in lubricating systems due to the considerably greater stability of purified oils (see Table 41).

Table 41. Results of comparative tests of oils.

Indices	AK-15 (avtol-18) based on GOST 1862-51	Nigrol Л based on GOST 542-50
Ratio of viscosities of scavenge and fresh oils		
on stand A	1,17—1,27	1,43
on stand B	1,34*	1,08**
Coking capacity (on stand B), %	1,71*	6,85**

Note: During bench tests A with toothed gears specific pressure at the pole is 14,900 kg/cm², rate of slip 0.64 m/s, duration of tests 150 h; on stand B — specific pressure at the pole 14,024 kg/cm², rate of slip 5.8 m/s.

*Duration of test 1000 h;
**Duration of test 500 h.

Thus, on the basis of base oils standardization of the assortment used can be carried out in the following way.

Table 42. Comparative appraisal of physico-chemical properties of scavenge transmission oils.

Indices	Nigrol based on GOST 542-50	Viscosine of mixture of oils
d_4^{20}	0,891	0,92
Kinematic viscosity at 100°, cSt	21,0	21,8
Ratio of viscosity at 50 and 100°	10,4	9,9
Index of viscosity	28	45
Coking capacity, %	3,2	0,8
Acid number, mg KOH	1,45	0,40
Deposit after oxidation at 163° for 48 h, %	3,56	2,29

Since the range viscosities, basically embracing the needs of lubricated industrial equipment, for lightly and moderately loaded mechanisms at 50° lies within the limits of 10-60 cSt, for liquid-friction bearings (application of high-quality highly purified oils of the aviation and turbine type) at 100° – 6-35 cSt and for heavily loaded mechanisms at 100° – 10-35 cSt, the following series of oils can be proposed for industrial equipment:

Series of oils	Nominal kinematic viscosity at	
	50°	100°
I-10	20	—
I-20	40	7
I-30	60	9
I-40	80	13
I-50	100	18
I-60	—	24
I-70	—	30

Maximum deviations from nominal values of viscosity can be within the limits allowed by the operational technical norms for commercial oils produced.

It is natural, that besides the considerations mentioned above stemming from the requirements of operational reliability and longevity of equipment, which are determined to a considerable degree by the quality of oils, it is necessary to consider the resources and cost of the oils recommended.

The data in Table 43 make it possible to conclude that from the point of view of lowering operational expenditures for the lubrication of industrial equipment the most acceptable sorts of oil are spindle, machine S, AK-10, AK-15, turbine, cylinder-2, 24, and 6.

Table 43. Cost of certain commercial oils.

Oil	Cost 1 t, rubles
Aviation oil MK-22	64—00
Bright stock P-28	64—50
Compressor T	58—00
[illegible]	58—40
As-6 with 10% SP-3	45—60
$AS_{3}p$-6	45—60
AK-10	20—80
AS p-10	45—60
$AK_{3}p$-10	80—90
$AS_{3}p$-10	45—60
AK-15	17—40
Spindle	14—50
Machine SU	45—60
Machine S	14—50
Compressor M	28—00
Motor T	16—20
Turbine	20—80
Transformer	43—30
MK-8	43—30
Cylinder-2 (11)	24—00
Cylinder-24 (viscosine)	11—60
Cylinder-6 (38)	24—00
Cylinder-52 (steam engine cylinder oil)	40—20

A comparative appraisal of resources of standardized oils with actual and future needs of enterprises which use the various sorts of oil will be subject to a thorough economic study.

Standardization of the Assortment of Oils for Industrial Equipment by Means of Selection of Rational Compositions of Additives

The assortment of oils for industrial equipment can be developed also on the base of the use of compositions of additives to one or several types of base (initial) oil. The basis of optimum compositions of additives should be an effective combination of polymeric additives and stabilizing additives. Possibilities were studied for obtaining a wide assortment of oils on the base of various sorts of commercial and test oils: AK-15 (based on TU 8-61); mazut deasphaltizer of Balakhansk oil; viscosine of selective purification; P-28 oil; transformer.

The mazut deasphaltizer of Balakhansk oil and viscosine of selective purification were obtained in the laboratory of oil technology of the INKhP of the Azerbaydzhan SSR Academy of Sciences.

Compounds obtained on the base of these products were appraised based on their viscosity and low-temperature properties. Figures 42-45 show the viscosity curves of mixtures of high- and low-viscosity components, and Table 44 and Figs. 46-49 — their pour point.

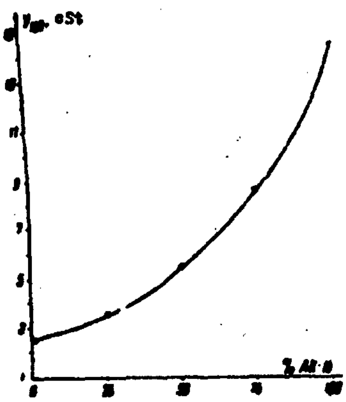

Fig. 42.

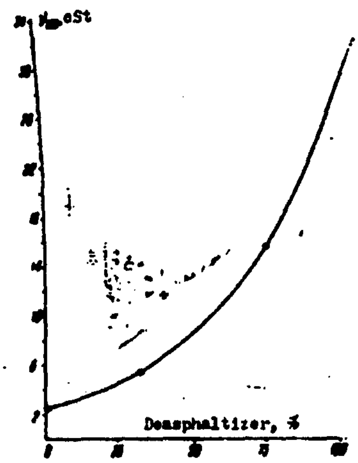

Fig. 43.

Fig. 42. Viscosity of compounds of transformer oil with AK-15.

Fig. 43. Viscosity of compounds of transformer oil with deasphaltizer of mazut BMN.

As can be seen, the same pour point (-30°) is ensured by a number of compounds:

25% transformer and 75% AK-15,
70% transformer and 30% deasphaltizer,
75% transformer and 25% viscosine,
65% transformer and 35% P-28.

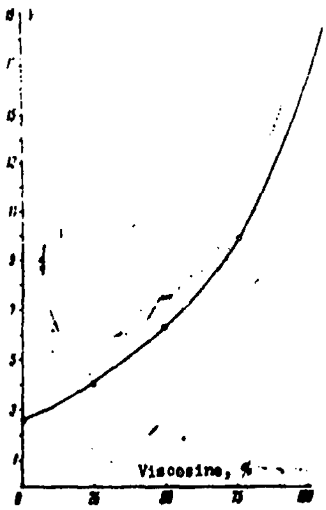

Fig. 44.

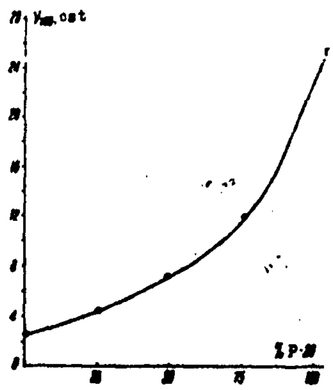

Fig. 45.

Fig. 44. Viscosity of compounds of transformer oil with viscosine of selective purification.

Fig. 45. Viscosity of compounds of transformer oil with P-28.

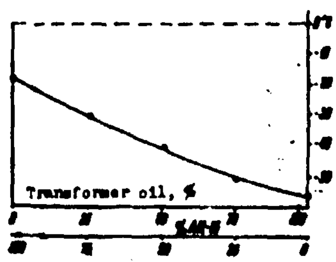

Fig. 46.

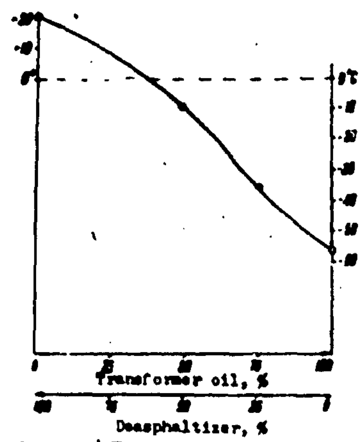

Fig. 47.

Fig. 46. Thickening of mixtures of AK-15 with transformer oil.

Fig. 47. Thickening of mixtures of transformer oil with deasphaltizer of mazut BMN.

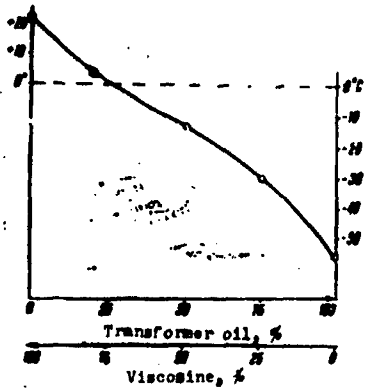

Fig. 48.

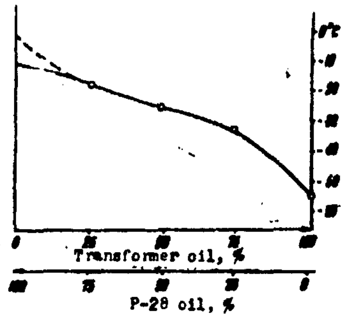

Fig. 49.

Fig. 48. Thickening of mixtures of transformer oil with viscosine of selective purification.

Fig. 49. Thickening of mixtures of P-28 oil with transformer oil.

Table 44. Pour point of initial oils and compounded oils.

Sample	Pour point, °C
AK-15 initial (TU AzSNKh No. 8-61)	−18
Mazut deasphaltizer of Balakhansk oil	+20
Viscosine of selective purification	+22
Transformer (GOST 982-56)	−56
Mixture of AK-15 with transformer	
25% + 75%	−50
50% + 50%	−40
75% + 25%	−30
Mixture of deasphaltizer with transformer	
25% + 75%	−36
50% + 50%	−10
75% + 25%	+10
Mixture of viscosine with transformer	
25% + 75%	−30
50% + 50%	−14
75% + 25%	− 4
Mixture of P-28 oil with transformer	
25% + 75%	−34
50% + 50%	−26
75% + 25%	−18

Certain basic indices of quality of the compounds are given in Tables 45 and 46.

Table 45. Corrosion of lead plates by the NAMI method (without catalyst).

Sample	Corrosion of lead plates after 25 h, g/m^2
AK-15 initial (TU AzSNKh No. 8-61)	120,2
Mazut deasphaltizer of Balakharsk oil	58,3
Viscosine of selective purification	50,5
Transformer (GOST 982-56)	+0,15
Mixture of AK-15 with transformer	
25% + 75%	99,05
50% + 50%	138,7
75% + 25%	150,9
Mixture of deasphaltizer with transformer	
25% + 75%	108,15
50% + 50%	71,9
75% + 25%	71,35
Mixture of viscosine with transformer	
25% + 75%	0,1
50% + 50%	0,65
75% + 25%	52,8
Mixture of P-28 oil with transformer	
25% + 75%	+0,25
50% + 50%	+0,25
75% + 25%	0,25

Table 46. Stability after oxidation in a DK-2 device (160°, 50 h)

Sample	Deposit after oxidation, %
AK-15 initial (TU AzSNKh No. 8-61)	11,4
Mazut deasphaltizer of Balakhansk oil	5,63
Viscosine of selective purification	6,56
Transformer (GOST 982-56)	2,01
Mixture of AK-15 with transformer	
25% + 75%	2,28
50% + 50%	6,56
75% + 25%	15,63
Mixture of deasphaltizer with transformer	
25% + 75%	7,18
50% + 50%	5,81
75% + 25%	6,71
Mixture of viscosine with transformer	
25% + 75%	3,5
50% + 50%	5,77
75% + 25%	9,25
Mixture of 25% P-28 oil + 75% transformer	4,17

The stability of oil, appraised by the deposit after oxidation at 160° for 50 h, for the normal operation of circulatory system used for the lubrication of mechanisms of industrial equipment, should probably comprise a value close to 3%. Proceeding from this, it is permissible to involve in the mixture up to 25% of viscous components

of type AK-15 and viscosine, and also up to 75% of P-28 oil. Anticorrosion properties of compounds obtained on a base of these relationships are satisfactory.

Turning to the dependences of viscosity of compounds on their composition (see Figs. 46-49), we see that in the case of involvement of 25% AK-15 it is possible to obtain a compound only with a viscosity of 3.7 cSt at 100°, and with 25% viscosine of selective purification – 4 cSt at 100°. A compound consisting of 75% P-28 oil and 25% of low-viscosity component gives a viscosity of 11 cSt at 100°.

Thus for the production of a wide assortment of oils the application of polymeric thickening additives is required.

However, the use of thickened oils for lubrication of mechanisms of industrial equipment is connected with a number of difficulties. In the first place this is stability agains destruction. The possibility of its control lies in the selection both of type of thickener, its optimum molecular weight, and concentration, and also of stabilizing additives.

A significant influence can be exerted here by antiwear and antiscoring additives, being found together in the solution with the thickener, inasmuch as the functional groups entering into the composition of these additives in combination with the thickening additives sometimes ensure a stabilizing effect.

In practice the problem of compilation of an assortment of oils in an assigned range is decided on the base of experimental curves, expressing the dependence of the necessary concentration of polymer with a certain known average molecular weight, necessary for obtaining the assigned level of viscosity. Figure 50 shows such curves, constructed for a case, when viscosity of the initial oil comprises 6.7 cSt at 100°. Analogous curves can also be obtained for any other initial level of viscosity.

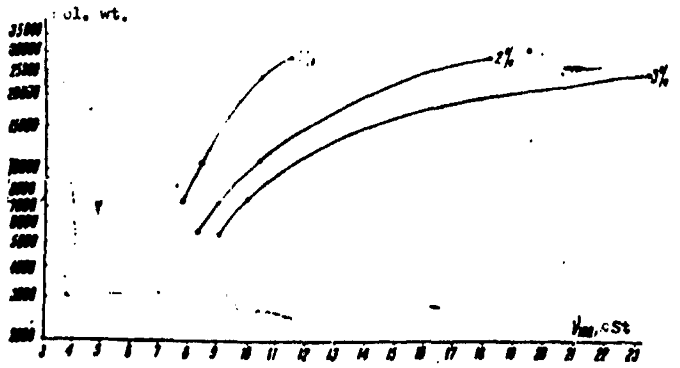

Fig. 50. Dependence of thickening capacity of polyisobutylene on average molecular weight.

For compounds having a viscosity of an order of 3.47 cSt (at 100°), as can be seen from Fig. 51, around 5% of commercial polyisobutylene of average molecular weight of 20 thousand is necessary in order to obtain oil possessing a viscosity of an order of 18 cSt at 100°. Viscosity-temperature characteristics of oils obtained are given in Table 47.

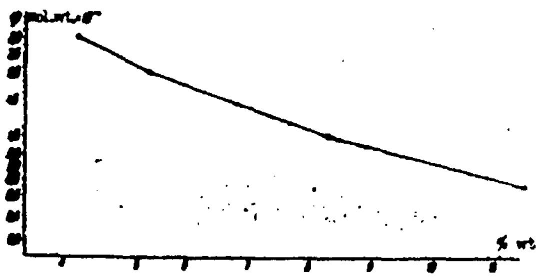

Fig. 51. Dependence of percentage of polyisobutylene in oil on the average molecular weight of the polymer.

Commercial polyisobutylene constitutes a mixture of fractions with a wide range of molecular weights. Fractions with a high molecular weight, as is known, are unstable to destruction. Therefore there is

interest in the thickening of a low-viscosity base by a fraction of polyisobutylene with a molecular weight of 10-11 thousand, the yield of which comprises 11.3%.

Table 47. Influence of polyisobutylene additives on the viscosity of oil.

Average molecular weight of polymer	Concentration of polymer in oil, %	Viscosity (cSt) at a temperature of (°C)											
		150	140	130	120	110	100	50	−0	−10	−20	−30	−40
30000	4,1	[illegible]	[illegible]	10,65	12,60	15,1	18,3	73,0	12,6	31	81	224	675
20300	[illegible]	7,88	9,12	10,60	12,60	15,1	18,3	[illegible]	14,7	38	105	316	1258
10000	8,2	7,65	8,90	10,56	12,55	15,2	18,3	81,1	16,6	48	162	630	2510
5000	12,0	7,70	8,88	10,64	12,62	15,1	18,3	84,9	26,2	68	218	890	3980
Initial oil		1,67	1,88	2,16	2,48	2,9	3,47	12,1	—	9	20	54	245

By thickening of a compound with a viscosity of 8.7 cSt with commercial polyisobutylene with a wide fractional composition (in a concentration of 2%) oil with a viscosity of 14.86 cSt at 100° and 84.14 cSt at 50° was obtained. However, the application of polyisobutylene as a thickening additive is hampered by its insufficient stability to depolymerization.

For lowering of destruction a copolymer synthesized by A. M. Kuliyev and L. M. Levshina was used. This was a copolymer of isobutylene with styrene (INKhP-20) which was characterized by a low average molecular weight (7800-9000). The additive of it to initial oil — a compound with a viscosity of 8.7 cSt — in a quantity 2.8% made it possible to obtain a mixture with a viscosity of 15.03 cSt at 100° and pour point of minus 32°. Index of viscosity of the product was 87.7.

In order to judge on the possibility of using the compounds obtained in mechanisms an investigation was made of their stability to destruction at sufficiently high rates of shift.

Preliminary tests on the appraisal of depolymerization stability of thickened oils were conducted on 1KhP installation for 4 h with a sampling of 4 tests of oil (after 15 min, 1, 2.5, and 4 h of operation).

As can be seen from the data in Table 48, the destruction of oil with polyisobutylene turns out to be 2 times deeper than with the copolymer. For the stabilization of destruction of solutions of polymers tests were made of various stabilizing additives. Table 48 contains data characterizing the ef... ...iveness of action of stabilizers. A decrease of relative destruction, observed with the addition of stabilizing additives to the solution of polymer, makes it possible to involve in the compound a greater percentage of low-viscosity component, which can considerably improve the properties of the oil.

Table 48. Relative destruction of solutions of polymers.

Sample	15 min	1 h	2.5 h	4 h
Spindle with 3% polyisobutylene	14,4	19,7	—	24,1
AK-10 + 5% copolymer, molecular weight 2800	8,24	11,35	—	12,65
AK-10 + 2.4% copolymer, molecular weight 7800	5.4	5,4	—	7,84
AK-15 with transformer (75 + 25%) + 1.9% commercial polyisobutylene	10,1	16,6	17,4	20,8
AK-15 with transformer (75 + 25%) + 2.8% copolymer, molecular weight 7800-9100	7,06	11,3	11,5	10,8
The same with 2% stabilizer 1	5 38	7,49	8,34	8,34
The same with 0.5% stabilizer 2	8,1	9,0	9,6	9.9

The experimental oil was tested under bench conditions. Results of the tests (Table 49, Fig. 52) testify to the high depolymerization stability of the sample with INKhP-20.

Table 49. Comparative appraisal of depolymerization stability of oils, thickened by polymers, on a stand with a closed circuit.

Time of sampling	Viscosity, kinematic at 50°, cSt		Loss of viscosity as a result of destruction (at 50°, cSt)	
	3% solution of copolymer (INKhP-20)	1.8% solution of commercial polyisobutylene	3% solution of copolymer (INKhP-20)	1.8% solution of commercial polyisobutylene
Prior to beginning of tests	95,43	95,35	—	—
After test				
15 min	94,56	93,34	0,87	2,01
24 h	89,15	85,93	5,41	7,42
48 h	84,67	79,38	10,76	15,97
72 h	[illegible]	[illegible]	12,98	19,51
96 h	81,00	72,63	14,43	22,72
120 h	78,00	70,42	17,43	24,93
144 h	78,00	69,16	17,43	26,19
168 h	76,39	68,78	18,64	26,57

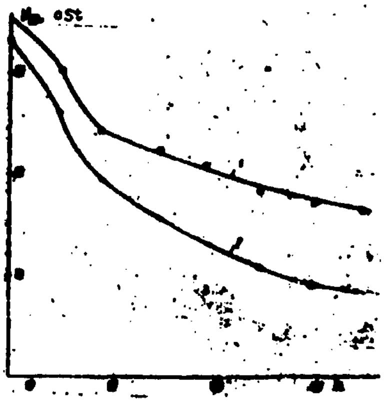

Fig. 52. Depolymerization of oils, thickened by various polymers. 1 – AK-15 + + transformer + 2.8% INKhP-20; 2 – AK-15 + + transformer + 1.1% polyisobutylene.

As can be seen, oil which is thickened with copolymer INKhP-20 is stabler to mechanical destruction under actual conditions of work of toothed gears. Destruction of oil, in which in addition to copolymer INKhP-20 antiwear and antiscoring additive INKhP-31 is introduced, is practically equivalent to the destruction of oil with the additive INKhP-20 (see Fig. 53).

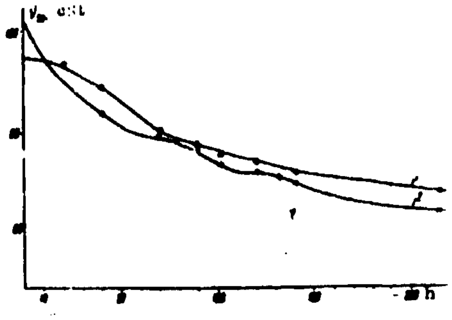

Fig. 53. Depolymerization of thickened oil with antiscoring and antiwear additive INKhP-31. 1 – test oil; 2 – test oil + 5% INKhP-31.

Thus, experimental data and experience of application show that

1) the thickening of low-viscosity compounds with the use of effective stabilizers can be used for obtaining a wide assortment of standardized oils for the lubrication of industrial equipment;

2) for the improvement of operational qualities of standardized oils utilized in mechanisms of industrial equipment, it is possible to be based on a complex of additives, of which the greatest interest lies in:

antipitting and antiwear – type INKhP-53, INKhP-54, AZA-3;

antiwear and antiscoring – type INKhP-16, INKhP-30, INKhP-31, INKhP-32, INKhP-46, INKhP-47, INKhP-50;

antioxidant – type AzNII-10, DF-11, AzNII-11f, AzNII-12, INKhP-38, INKhP-35, INKhP-36, INKhP-25;

thickening – type INKhP-20;

antifoam – polymethylsiloxanes and others.

CHAPTER VII

PROSPECTS OF USING ADDITIVES TO LUBRICATING OILS FOR DRILLING EQUIPMENT

The five-year plan for the development of the national economy of the USSR during 1965-1970 provides for a considerable increase in the output of oil and gas, for which a great deal of attention is allotted to expansion of the volume of prospecting and operational drilling (by 1970 it is planned to carry out more than 43 million meters of operational and almost 62 million meters of prospecting drilling).

In the practice of prospecting and operational drilling the most widespread methods are those of rotary drilling (rotary and turbine, and also with electric drills). All drilling rigs for rotary drilling of deep wells consist of the following basic groups of equipment: rotary, ensuring rotation of the boring tool; lifting – for raising from the wells and lowering into them of drilling pipes and tool, retention in a suspended state, and lowering of casing columns in the well; pumping – for pumping washing liquid into the wells; power, putting into motion the units and mechanisms of drilling equipment.

A diagram of the interaction of mechanisms of drilling equipment in the process of rotary drilling is shown in Fig. 54. If driving of the operating unit (bit) is carried out from motors mounted on the surface (during rotary drilling), the flushing liquid being pumped

through the well, in passing the bit reaches the face and from there along the pipe space together with the drilled rock reaches the surface. During drilling with a turbodrill the flushing liquid passes through the turbodrill and causes its rotation. In the rotary method of drilling the bit revolves with the help of the column of drilling pipes.

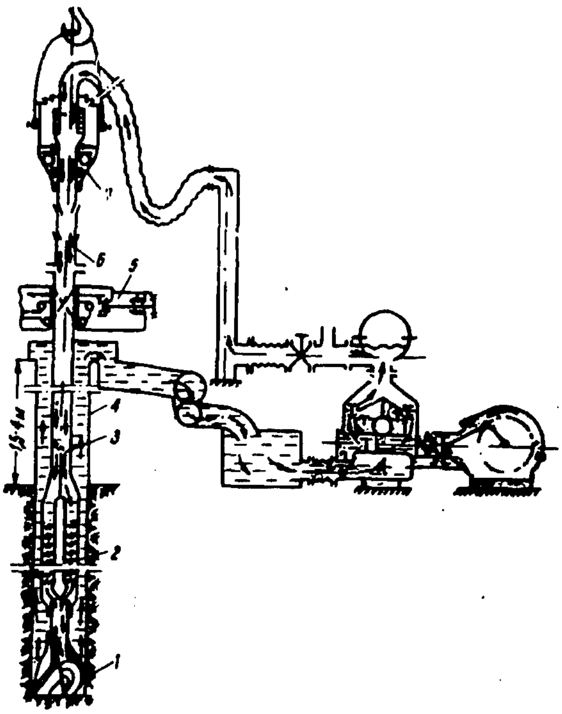

Fig. 54. Diagram of the interaction of mechanisms for drilling. 1 – bit; 2 – face engine or loaded bottom; 3 – column of drilling pipes; 4 – casing column; 5 – rotor; 6 – driving bar; 7 – swivel.

For separating the rotating part of the drilling tool from the nonrotating there is a swivel. The set of drilling equipment also includes mechanisms for power driving, mechanisms to ensure circulation and purification of flushing liquid, for lowering and hoisting

operations, and other auxiliary devices. If an electric drill is used as the face motor energy transfer to the face is carried out with a special current feeder.

In process of drilling rotation of the face instrument is combined with its gradual motion onto the face. Simultaneously the carrying away of drilled rock is ensured with the help of the flushing liquid or gas.

The effectiveness of the process of sinking of wells is combined in many respects with the reliable operation of the equipment and depends basically on the period of service of the bit on the face. The latter is conditioned to a considerable degree by the depth of sinking. With an increase of depth the average sinking decreases, and accordingly the value of average mechanical rate of drilling decreases.

Inasmuch as productivity of the drilling process depends in many respects on driving of the bit and change in the mechanical rate of drilling, the value of which is reduced sharply with depth, it is natural that intensification of the drilling process can proceed both by way of increasing mechanical speeds, and also by increasing the driving of the bit. In the case of drilling wells by the rotary method the mechanical rate can be increased at the expense of increasing the load on the bit. Intensification of drilling at the expense of increasing the drilling of the bit when drilling with face motors (turbodrills and electric drills) can be achieved by means of increasing the service period of bits. With an increase of depths of drilling, by decreasing the diameter of the wells and the application of turbodrills the powers of driving motors and rates of rotation of the bit are increased sharply.

Depending on the construction of the bit they are subdivided into cutting and blade; and according to the nature of influence on rocks – crushing-shearing, cutting, and abrasive cutting. Furthermore, rock can be destroyed by abrasion with the application of diamond bits.

Based on assignment of bit for rotary drilling they are divided into bits for solid destruction of the face, column, and those of special assignment.

Contemporary designs of bits used for sinking of wells are mainly cutting bits. When they are used in turbine drilling rates of rotation up to 500-600 r/min are realized, and during rotary drilling — up to 350 r/min.

The most widespread are three-cutter bits. Each of the cutters constitutes a construction, including support, rolling elements, and the cutter itself. The location of the support elements of cutter bits can be very diverse (Fig. 55). For bits which include ball elements component races are characteristic.

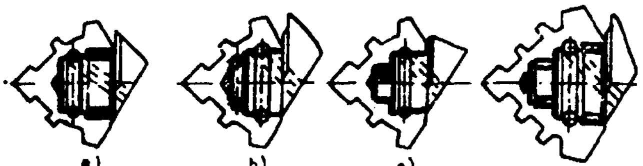

Fig. 55. Constructional diagram of supports of cutting bits. a) double-seat (ball and roller); b) the same (double-ball); c) the same (ball with sliding support); d) three-support (roller — ball — roller).

Analysis of numerous experimental data shows that in the breakdown of supports of bits the leading form of breakdown is contact fatigue breakdown of working surfaces of the journal races, and also of the balls. It is namely on the balls and journal races where the first criteria of breakdowns usually appear. Initial roughness of surface changes considerably, and separate foci of pitting breakdowns appear, leading to pitting and scaling of surface layers of metal.

Before the most intense phase of breakdown of support surfaces, by which the finish of the bit is determined, fatigue breakdown is already taking place on all the surfaces (a considerable change of linear dimensions of supports leads to misalignment of rollers, axial shift of cutters, contact of faces of cutters with the ends of journal drills, and, as a rule, to wedging of cutters).

Thus, the most important factor for increasing the service period of a support is increasing the duration of the initial period.

The addition of additives to flushing solutions and to lubricating compositions, which are usually used for improving the conditions of work of bit supports (both hermetically sealed and also unsealed), is an effective means of increasing their period of service.

For exposing the effectiveness of the influence of additives on the development of contact breakdowns various test methods can be used [196-205].

For this purpose we used a unit with three balls. Operating conditions of the tests are determined proceeding from the real values of loads and speeds on working elements of the bit support (see Fig. 56).

Loads on support surfaces, the value of which will determine limit of contact fatigue, are established from the following dependences [206]:

$$Q_1 = \frac{P}{b}\sqrt{1+\left(\frac{kD_1}{D_{\text{ш}1}}\right)^2}\,(a_1\cos\alpha - D_{\text{ш}1}\sin\alpha) + \frac{TD_{\text{ц}}}{2b},$$

$$Q_2 = \frac{P}{b}\sqrt{1+\left(\frac{kD_1}{D_{\text{ш}1}}\right)^2}\,(a_1\cos\alpha + D_{\text{ш}1}\sin\alpha) - \frac{TD_{\text{ц}}}{2b},$$

$$N = P\sin\alpha + \frac{P_{\text{г}}}{\sin\alpha},$$

[illegible] – longitudinal force acting on the cutter;

[illegible] $= \frac{1}{3}$ [illegible]$_{общ}$ – total load on bit, kgf;

k – coefficient of resistance to rotation of cutters on face, (its value changes strongly depending on the characteristics of the rocks, type of bit, and its state:

for bits of type C and T when sinking through hard and moderately hard rocks k = 0.1-0.2; when sinking through soft rocks k = 0.2-0.3; for strongly worn out bits k = 0.4-0.45);

D_1 – diameter of face ring;

$D_{ш1}$ – diameter of rims of cutter, cm;

a_1 – distance from internal support to point of application of load, cm;

$D_м$ – diameter of cutter, cm;

$P_д$ – load of forces of inertia, developing on the cutter during rotation of the bit:

$$P_д = \frac{G \omega^2 R}{2g},$$

where G – weight of cutter, kgf; ω – angular velocity of rotation of bit

$$\omega = \frac{\pi n}{0}$$

R – average radius of bit, on which the center of gravity of the cutter is located

$$R = \frac{D_{ср}}{2} = \frac{D}{6}.$$

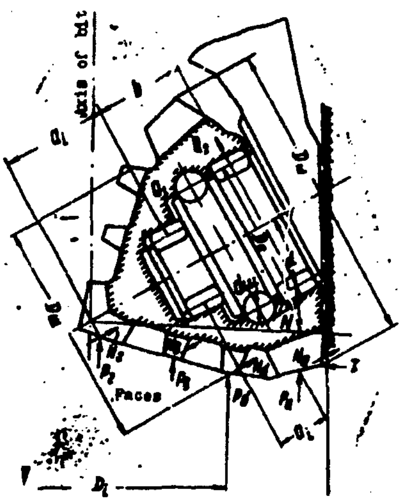

Fig. 56. Loads, having an effect on the support elements of a bit.

Results of preliminary tests, conducted at the IKhP of the Azerbaydzhan SSR Academy of Sciences, made it possible to establish that when phosphorus-, nitrogen- and sulfur-containing compounds are used as additives it is possible to increase significantly the stability of ball elements.

The power drive of drilling rigs, autonomous in particular (diesel, diesel-electric, or gas turbine), is realized mainly by (DVS) [internal combustion engine] of serial types with mechanical and hydraulic transmissions.

During rotary drilling the main power is consumed by pumps and the rotary table, but during lowering and hoisting operations — by the winch; when drilling by the turbine method the main share of power is consumed by the pumps.

The operational reliability of the power drive has a significant influence on the most important indices of drilling. The application

of a DVS with turbotransformers ensures a great flexibility of engine performance. As can be seen from the data in Table 50, inclusion of a hydraulic transmission in the kinematic arrangement makes it possible to essentially change the relationship between maximum M_{max} and nominal $M_{н}$ torque, and also the ratio of maximum rate of rotation of the engine n_{max} to minimum n_{min}.

Table 50. Flexibility of characteristics of various kinematic layouts for driving drilling rigs.

Characteristics of drive	$\frac{M_{max}}{M_{н}}$	$\frac{n_{max}}{n_{min}}$
DVS of average speed (500-750 r/min)	1,1–1,15	1,5–2
DVS with increased speed (1200-1500 r/min)	1,0–1,1	1,3–1,8
DVS with torque converter	1,5–3 5	1,5–3
DVS with electric power transmission (direct current)	1,6–2,2	2,5 4,0
Asynchronous electric motor	1,7–1,0	1,0
The same, synchronous	1,65	1,0

Specific weight of the electric drive of drilling rigs comprises 40%, and installations with a drive made up of a DVS with mechanical and hydraulic transmissions – around 60% of the overall number of installations. Gas turbine engines are also used as the power drive of drilling rigs. They possess good response, sufficiently high reliability, and satisfactory adjustability (coefficient of multiplicity of torque). Although the effective efficiency factor of a gas turbine unit (0.12-0.27) is considerably lower than that for a DVS (0.36-0.38), the use of gas turbine units is promising, inasmuch as their efficiency can be increased by using free-piston engines.

As the power drive of the DVS the widest use is made of diesel engines with the dimensions 12ChCh 15/18 (12ChN 15/18) and 12ChN 18/20. Brief characteristics of them are given in Table 51.

Table 51. Technical characteristic of engines.

Indices	12Ch 15/18	12ChN 18/20
Effective power, horsepower	300	1000
Number of turns, r/min	1500	1500
Average effective pressure, kg/cm²	4.7	9.3
Specific effective expenditure, g/e.h.p.h		165
fuel	175	
oil	9	8
Capacity of lubrication system	40–75	75–30
Period of service of engine, h		
up to first overhaul	2000	—
up to major overhaul	5000	6000

The influence of alloying of lubricating oils on improvement of operational indices of a DVS is well-known. This question has been treated by numerous investigations, a considerable share of which are being conducted at the IKhP of the Azerbaydzhan SSR Academy of Sciences (earlier at the INKhP AN Azerb. SSR and the AzNII NP) under the leadership of A. M. Kuliyev [207].

The basic operational factor which is actively influenced by alloying of the DVS lubricant is its longevity, determined by the wear of basic components. It is known that the use of effective additives [95] can increase the period of service of an engine by 1.5-2 times [188]. A significant effect during the exploitation of DVS on oils with additives is attained with an increase in the periods of service of the oil in them. Thus, according to V. A. Somov [208], an increase in the period of service of oil by 2 times ensures a saving of 25 rubles per 1000 h of operation for the ZD12 (12Ch 15/18) engine and 17.5 rubles per 1000 h of operation for the M-756 (12ChN 18/20) engine.

<u>Hydraulic transmissions</u>. The use of hydraulic transmissions in drilling, besides the described advantages in the stage of ensuring satisfactory indices of flexibility of characteristics of the power drive of drilling rigs, also permits the carrying out of shockless inclusion of load during operation, which excludes the action of dynamic loads on driving elements and, mainly, ensures the stepless

control of speed of servomechanisms. Under these conditions it turns out to be possible to support a constant value of power during a change of loads. Hydraulic transmissions, used widely in the kinematic circuits of contemporary power drives of drilling rigs, include basically fluid couplings and hydraulic converters.

Conditions of work of working fluids in hydraulic transmissions, where chiefly mineral oils are used, have to ensure the reliable operation of the system in a wide range of operating temperatures. High rates of flows of oil in the circulation system, and also continuous admission of air promote the intense heating of oil, its oxidation, and varnish formation.

The use of low-viscosity working fluids promotes increase of efficiency of hydraulic converters, however, the appearance of cavitational conditions is possible.

The reliability of operation of a system of hydraulic transmission and especially of automatic control systems is influenced considerably by the viscosity-temperature characteristics of the oils used as the working under conditions of friction, rolling and slip (gears, free wheeling clutches, bearings, and others), longevity with antiscoring, antiwear, and antifriction properties is imparted to it, as is known, by a special complex of additives.

An important condition for ensuring the reliable performance of hydraulic transmission is the satisfactory anticorrosion characteristic of the oil used as the working fluid with respect to aluminum and magnesium alloys, from which the working wheels and bodies are frequently made, and also the various inserts and support washers (material – basically alloys on a copper base). The danger of formation of varnish deposits, precipitates, and slimes under conditions of intense oxidation of the oil presents special requirements for thermooxidizing stability of the oil, which is effectively improved by various additives.

When low-viscosity lubricating oils with thickening additives (polyisobutylene, Vinypol, copolymers of isobutylene with styrene, and others) are used as working fluids for a hydraulic drive it is necessary that they have sufficient stability agains depolymerization. The stability of thickened oils to depolymerization phenomena is increased effectively by the introduction of stabilizers [207].

In contemporary hydraulic transmissions complex compositions of oil additives are used, thus increasing the reliability of operation of hydraulic converters. Thus, VNII NP-1 oil (GOST 10660-63) constitutes a thickened polyisobutylene low-viscosity dewaxed oil, to which dialkyl dithiophosphoric DF-1 additive, the antioxidizer phenyl-α-naphthylamine, and an antifoam additive are also added.

Transmissions. In the power drive of a drilling rig gear boxes are used widely for changing the number of turns of servomechanisms. Their period of service, determined by wear of the gears, sprockets of chains, should be no lower than the period of service of the remaining basic units of the drilling rig, for example, the rotor of winch. Proceeding from this, increasing the efficiency of the basic wearing elements of transmissions with the help of additives is a very effective method. Furthermore, with an increase of load capacity (for example, of the gears) it is possible to ensure a decrease in the dimensions of transmissions, which has a positive effect on vibration resistance and dynamic loads in the drive.

Toothed gears usually include double-helical and skew bevel gears.

When a transmission is realized exclusively with toothed gears the leading forms of maximum states are fatigue breakdown from the action of contact stresses, wear of tooth profiles, breakdown of teeth, also having basically a fatigue nature, and jamming [209-221].

The numerous constructions of transmissions for drilling rigs include chain elements. Breakdowns here also are as a rule of a fatigue nature. Prior to repair transmissions should ensure the drilling of no less than 50-80 thousand tons. By the addition alloying components to the lubricant it is possible between transmission repair periods to reach 100-120 thousand tons of drilling.

In the drive of drilling rigs systems are also used where toothed gears are used simultaneously with a chain drive. This complicates the selection of additives for increasing the service period of transmissions, inasmuch as experience in the application of certain effective additives, improving a number of the most important operational qualities of oils for toothed gears (antiscoring and antiwear), shows that these additives negatively affect the resistance of elements of kinematic pairs to breakdowns due to contact fatigue. A characteristic exampl of such an additive is chloroparaffin. According to Scott [201], the introduction of chloroparaffin reduces the time up to the beginning of formation of pitting on steel surfaces, whereas the generalized index of wear, determined by GOST 9490-60, for the same concentrations of this additive increases sharply. Meanwhile it turns out to be possible to select such a combination of antipitting, antiwear and antiscoring properties of a lubricating composition, utilized in a particular mechanism, that the efficiency of its basic elements, determined by the different nature of maximum states, will be equivalent.

<u>Swivels</u> are an intermediate link between the forward moving tackle system and the drilling sleeve with the rotating drilling tool. The basic constructional layout of a swivel is shown in Fig. 57.

As shown by the results of special investigations, conducted by the VNIIPTNYeFTYeMASh [209], the basic cause of breakdown of swivels is the insufficient service period of bearings of the main supports. S. G. Babayev and I. V. Shlimak [209] inspected U6-ShV14-160M swivels in the process of overhaul. As basic support in this construction of a swivel a No. 19742 thrust bearing with conical rollers is used.

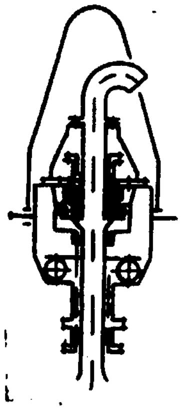

Fig. 57. Constructional layout of a swivel.

According to this inspection the greatest number of defects, causing the breakdown of the main support, pertain to damages of beads (51%), and 44% of breakdowns the authors connect with fatigue breakdowns due to pitting wear. Breaking off of beads is the result of blows of the rollers on them from radial loads which develop during rocking of the drilling column, and elimination of this defect in no way can be connected with a change in the quality of oil used if one does not consider the possibility of intensification of radial displacements of the shaft in the case of wear of radial bearings.

Breakdowns of the support surface, having the nature of fatigue, are effectively eliminated by the addition of AzA-3 additive to the lubricant. Figure 58 shows photographs, characterizing the state of swivel supports during work on oil without additive and with additive. In ShV-15-300 swivels, where thrust bearings with cylindrical rollers are used, fatigue breakdowns of working surfaces appear more distinctly. Furthermore, the construction of the support of this swivel does not anticipate limiting beads on the radial bearing. Due to this main share of breakdowns here are progressing crumbling and pitting wear.

<u>Rotors</u> are typical reduction gears with crossing axes. The transmission of rotation from the horizontal drive shaft vertically to the suspended column of drilling pipes is carried out by bevel gearing.

The basic layout of constructional performance of rotors is a single-stage transmission of rotation from the drive shaft onto the table of the rotor. Designs of two-stage reduction gears are known where bevel gear is intermediate and the drive mechanism of the table is carried out by cylindrical gears.

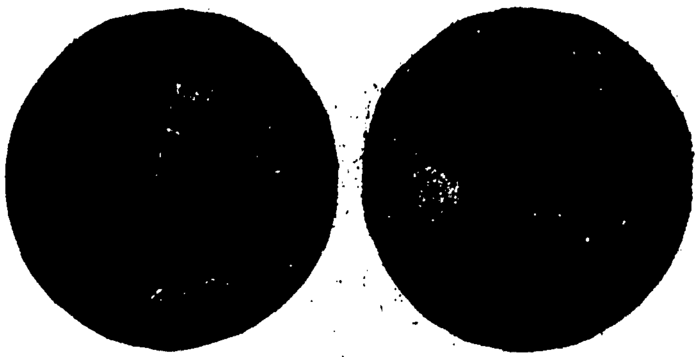

Fig. 58. State of the surface of the main support of a swivel during work on industrial-45 oil without an additive (a) and with 4.2% AzA-3 additive (b).

As S. G. Babayev and I. V. Shlimak point out [209], the basic cause of loss of efficiency of rotors is breakdown of the main support, where in this case progressing breakdown due to contact fatigue predominates. These authors note that from the R 560-Sh8 and 47-520-3 rotors inspected by them in 77 and 90% of the cases respectively contamination of the lubricating oil with clay solution was noted. An investigation of the influence of clay solution on breakdowns due to contact fatigue was conducted by E. G. Kister, R. S. Lerner, and G. V. Rogozina [205]. It is also known [116] that the introduction of water into lubricating oil sharply lowers the time before the beginning of fatigue breakdowns.

At the same time it has been established [117, 118] that by the introduction of additives into the lubricating medium, where the water which causes pitting formation is present it is possible to neutralize

its harmful action. Such additives are isoamyl alcohol, dehydrating compositions, derivatives of imidazolines, and also compositions of them.

The influence of many surface active additives, the mechanism of action of which is connected with facilitation of emergence of dislocations onto the surface, perhaps also is caused by a change in the resistance of the material to breakdowns due to contact fatigue. Breakdowns of the working surfaces of teeth of the conical pair of the rotor are also basically caused by contact fatigue, where for the teeth of straight-toothed gears of the R 560-8Sh rotor the specific share of these breakdowns is higher than for the teeth of the helical transmission of the 47-520-3 rotor.

Thus the longevity of a conical pair, to the same degree as this takes place for support bearings, depends on the influence of additives on a change in the resistance of surface material to breakdowns due to contact fatigue.

Reduction-gear electric drills. For increasing the value of torque, transmitted to the face tool during drilling, it is necessary to reduce the number of revolutions of the face motor. In general the most favorable characteristic of adaptability of torque, as was shown above, perhaps is attained with the inclusion, in the kinematic system from the face motor to the face tool, of a stepless variator of the hydraulic-converter type. However, with a single-stage transformer of this type the diameter of working wheels would be too great, and a satisfactory constructional solution for multistage hydraulic converters has still not been found.

The reduction gears used for reducing the number of revolutions of the electric face motor are installed within the dimensions of the electric drill and constitute a multistaged gear. Such, for example, is the construction of the Soviet reduction-gear electric drill of

the type ER-215, designed by the Central Design-Technological Bureau of Electric Drilling (TsKTBE). The reduction-gear electric drill of this type is made on the base of the serial E-215 electric drill. The kinematic layout of the ER-215 reduction-gear electric drill includes two stages – cylindrical and conical. In both cases the gear is of the planetary type, where in the cylindrical stage the satellite gear has needle bearings as supports, and in the conical stage – cylindrical ball. Brief technical characteristics of the ER-215 reduction-gear electric drill are:

Power	100 kVh
Voltage nominal	1100 V
Current	86 A
Rate of rotation of output shaft	209 r/min
Moment nominal on output shaft	400 kgm
Nominal diameter	220 mm
Length	
full	10,726 mm
working	1023 mm

Constructional shaping of the reduction-gear electric drill is such that as the lubricant for the gears of both stages a low-viscosity oil, filling the cavity of electric motor, is used. Probably due to this during the process of work of the reduction-gear electric drill frequently the small satellite gears of the cylindrical stage go out. The maximum state here, as their testing showed, is jamming of working sections of the tooth profile. The high temperatures appearing on friction surfaces lead to local breakdowns and, as a result, sometimes to breakdown of the gear.

An increase in the limits of efficiency of the toothed gearing of the cylindrical stage can be attained by adding effective antiscoring and antifriction additive to the oil [222].

The above analysis of basic trends in the alloying of technical oils with the help of multifunctional additives shows that in the most diverse conditions of their application a rational approach to

the selection of additive ensures the obtaining of a significant effect, amounting to a considerable increase in the reliability of the equipment.

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UNCLASSIFIED
Security Classification

DOCUMENT CONTROL DATA - R & D

(Security classification of title, body of abstract and indexing annotation must be entered when the overall report is classified)

1. ORIGINATING ACTIVITY (Corporate author)
Foreign Technology Division
Air Force Systems Command
U. S. Air Force

2a. REPORT SECURITY CLASSIFICATION
UNCLASSIFIED

2b. GROUP

3. REPORT TITLE
USE OF ADDITIVES FOR LUBRICATION OF INDUSTRIAL EQUIPMENT

4. DESCRIPTIVE NOTES (Type of report and inclusive dates)
Translation

5. AUTHOR(S) (First name, middle initial, last name)
Kuliyev, A. M., Suleymanova, F. G. and El'ovich, I. I.

6. REPORT DATE
1969

7a. TOTAL NO. OF PAGES
206

7b. NO. OF REFS

8a. CONTRACT OR GRANT NO.

b. PROJECT NO. 72302-78

c.

d.

9a. ORIGINATOR'S REPORT NUMBER(S)
FTD-MT-24-89-70

9b. OTHER REPORT NO(S) (Any other numbers that may be assigned this report)

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11. SUPPLEMENTARY NOTES

12. SPONSORING MILITARY ACTIVITY
Foreign Technology Division
Wright-Patterson AFB, Ohio

13. ABSTRACT

The development of research works in the field of synthesis and production of oils additives created conditions for the radical improving of quality of lubricating oils and made it possible to change these qualities in the required directions. The book covers the basic principles for determination of the necessary level of alloying, proceeding from an analysis of conditions of operation of the basic forms of industrial equipment and their requirements for the quality of additives for lubricating oils. For a rational organization of investigations in the field of selection of additives for the basic types of industrial equipment the machinery and mechanisms are classified by structural, kinematic, and dynamic factors. An analysis of efficiency based on maximum states of kinematic pairs makes it possible to determine the basic directions and required level of alloying under specific conditions. With the help of the complex of preliminary methods of sampling and tests of additives on laboratory machines, instruments, and installations, including an appraisal of functional properties of additives to oils (antiwear, antipitting, antiburr properties, stability, corrosion, depolymerized stability and others), and also a complex of methods of bench tests, the appraisal of the most important operational qualities of additives is ensured.

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14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Lubricant Additive Industrial Equipment Lubricating Oil Drilling Equipment						

211

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